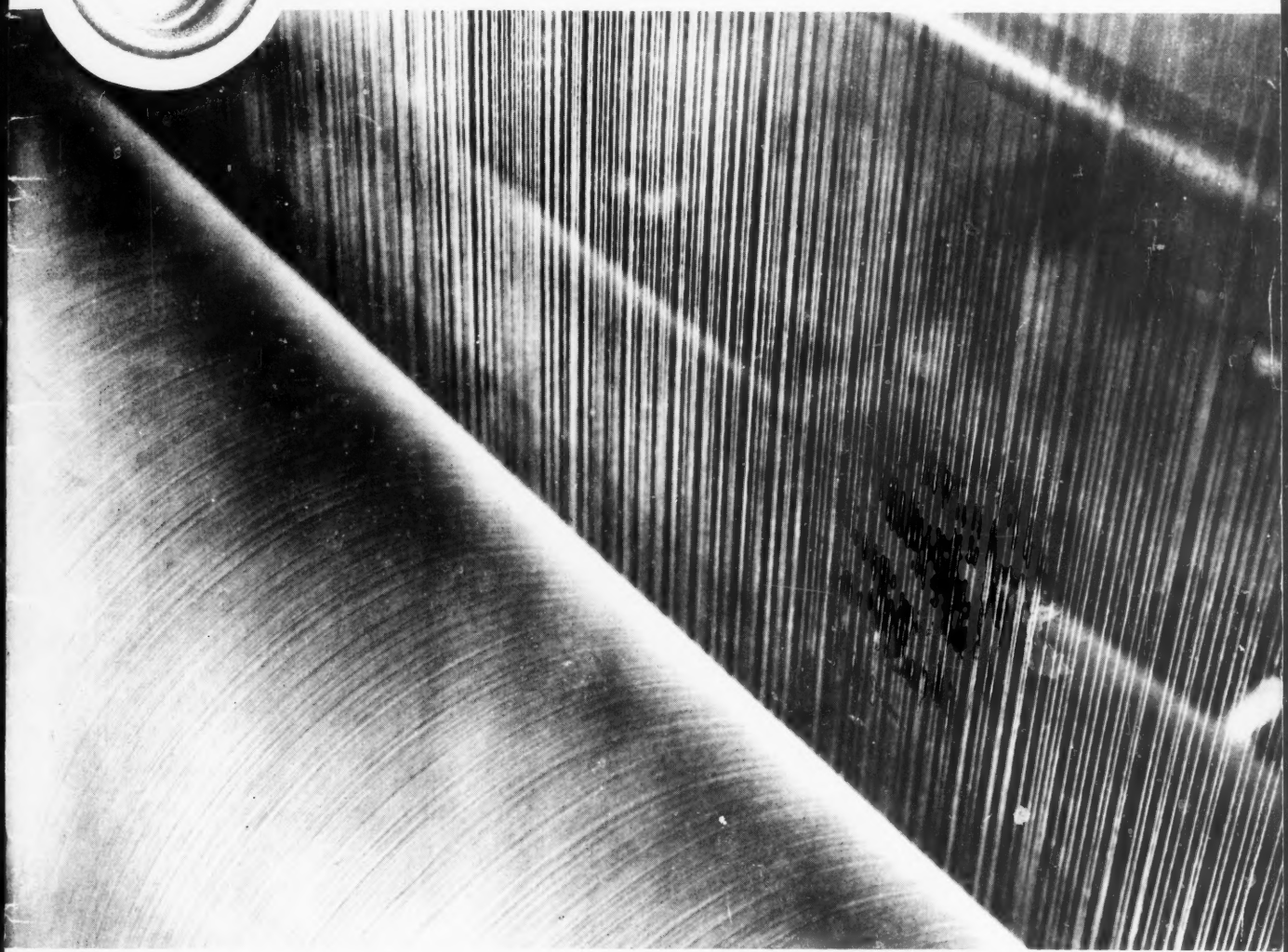
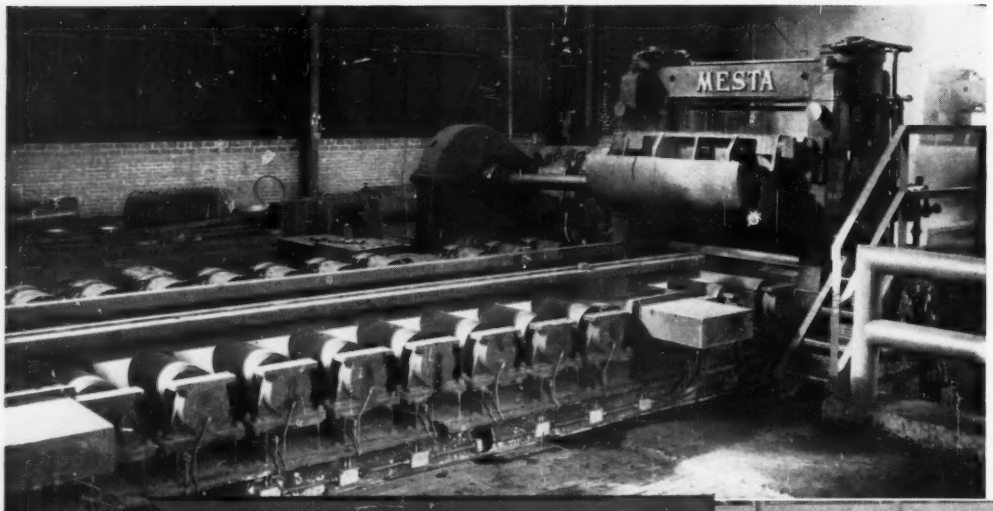
FEBRUARY, 1954
Vol. 10, No. 1

Lubrication ENGINEERING



Practical Feature: TEXTILE MILL LUBRICATION

JOURNAL OF
THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS



260-point, newly designed, run-out table at Allegheny Ludlum, automatically lubricated by Trabon.

11 bearings on the revolving mandrel of this Allegheny Ludlum coiler are quickly and easily lubricated by Trabon.

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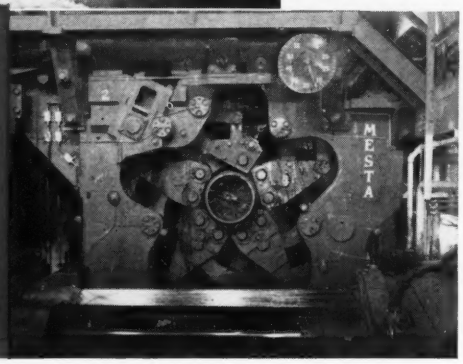

this has a BEARING on the subject

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There are good reasons for Allegheny Ludlum's choice of Trabon automatic lubrication systems:

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- It's completely sealed so that it operates effectively even underground and is unaffected by steam, dirt or snow.
- Trabon systems, for either oil or grease, fit any bearing situation.

Write for more information today.

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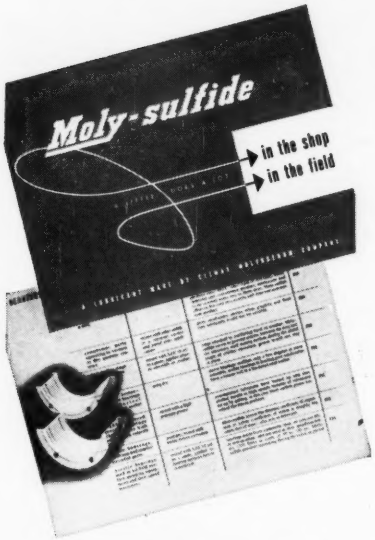
154 ideas on ways to use...



154 varied applications of molybdenum sulfide in the shop and in the field are described in a new booklet now available. This solid-film lubricant has demonstrated unique anti-friction properties under conditions of extreme pressure, high velocity, elevated temperature, or chemical attack.

The 40-page booklet contains the records of solved lubrication problems — some might solve your own. Fill in the coupon below, attach it to your letterhead and send it off today.

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Lubrication ENGINEERING

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(Cover Picture — Kaufmann & Fabry Photo)

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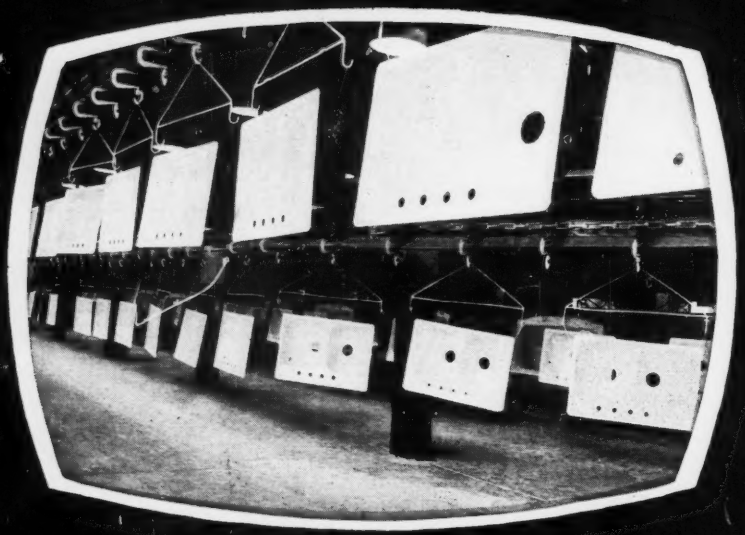
JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

telling the story of 'dag' dispersions



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tests prove

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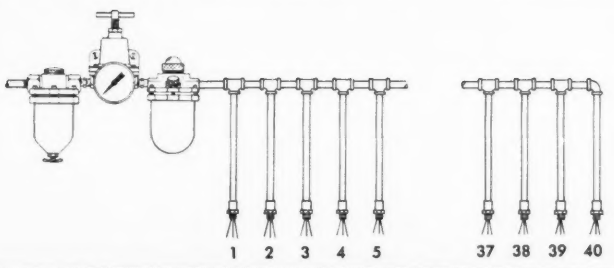

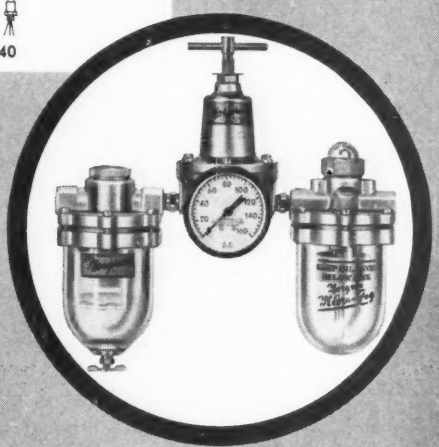

The extremely fine air-borne Micro-Fog provides absolutely uniform distribution of lubricant through as many as 40 outlets... for bearings, spindles, gear boxes, etc.

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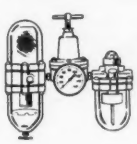
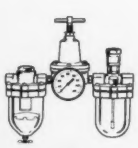


Micro-Fog Lubro-Control Unit for bearings, spindles, gear boxes.



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Micro-Fog Lubricator, tank type, 1¾ gal. cap.

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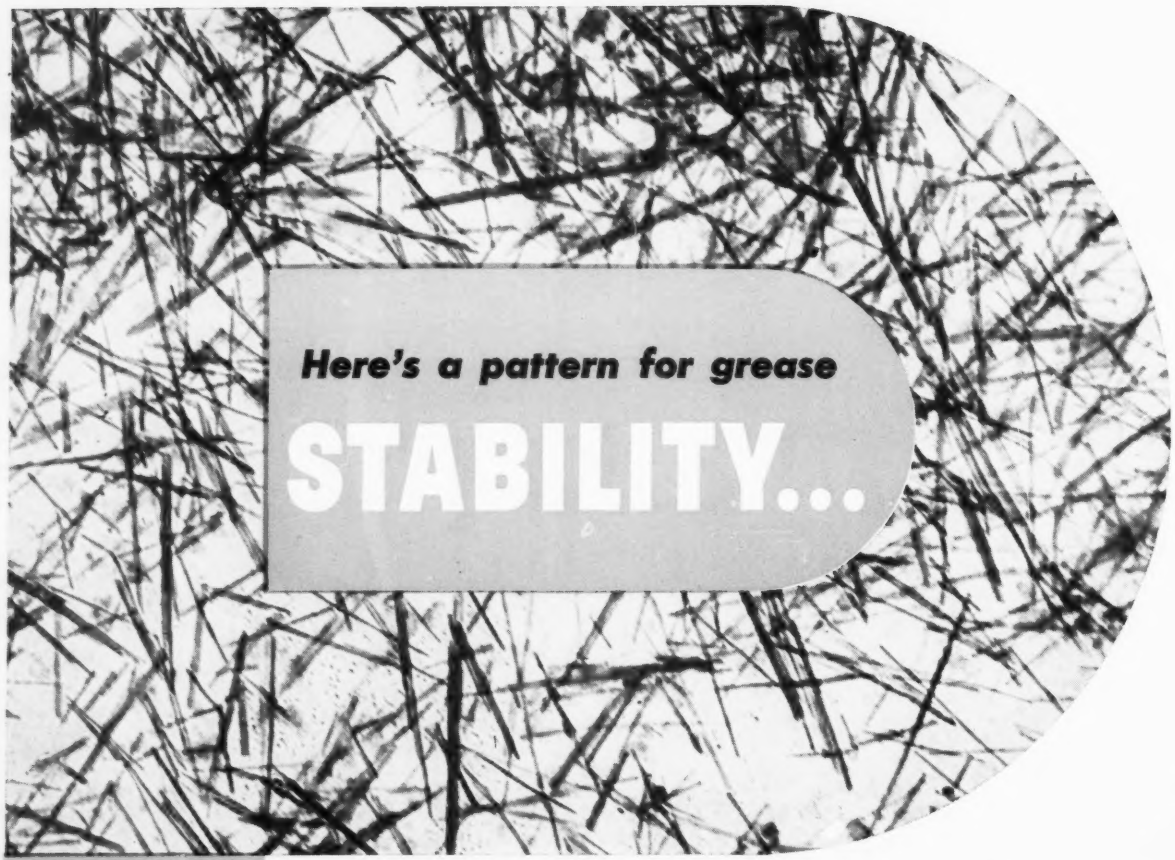

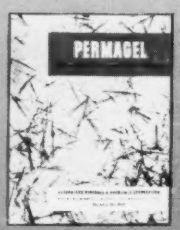

Here's Help . . .

Ask for Bulletin P-53. It discusses Permagel's properties, and describes in detail procedures, equipment and formulation factors involved in making Permagel greases.

The colloidal network pictured here is Permagel, the pure inorganic mineral attapulgite. Dispersed in lubricating oils, these needle-like particles result in multi-purpose greases that are both stable and highly resistant to mechanical shear.

What's more, greases thickened with Permagel *remain* stable under a wide variety of operating conditions. They maintain original consistency over a broad range of temperatures and in high shear service as well.

Outstanding mechanical stability is just one of Permagel's many helpful properties. Permagel multi-purpose greases have excellent low temperature pumpability, no melting or dropping points, and show excellent physical stability (because an inorganic agent does the gelling). They have excellent water washing resistance and can absorb more than their own weight in water with little change in consistency.

Permagel's uniformity from shipment to shipment practically insures grease reproducibility. You can select the surfactant best suited to your needs, thus allowing you wide process flexibility. And, all of Permagel's outstanding properties are yours at *unusually low cost.*

Want more information? We'll be happy to forward you detailed technical data and generous samples of Permagel for your evaluation. Or maybe you'd like us to process your oil in our modern grease pilot plant. In either case our technical knowledge is yours for the asking.

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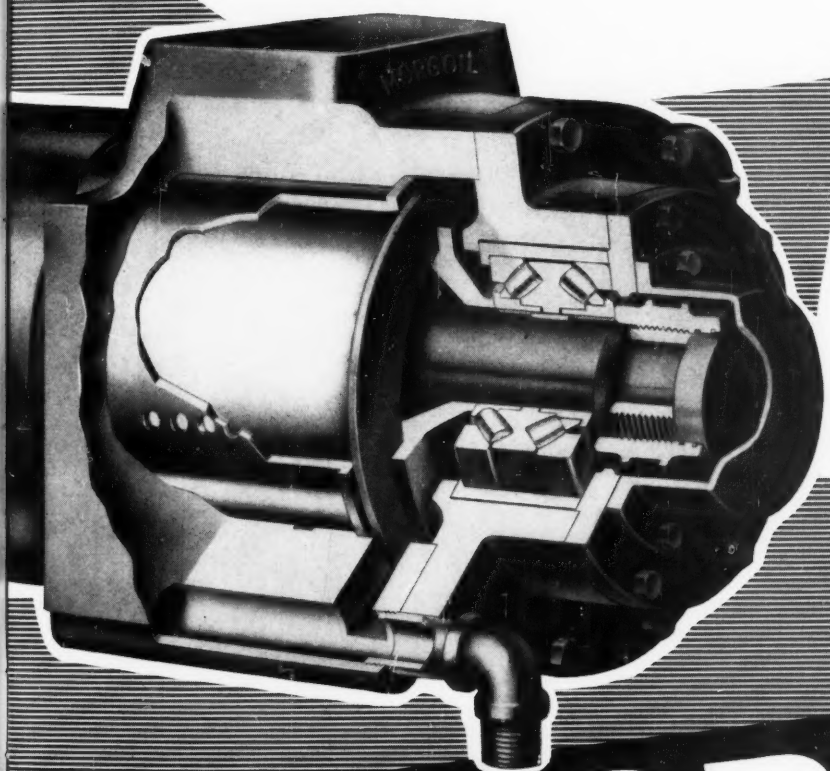

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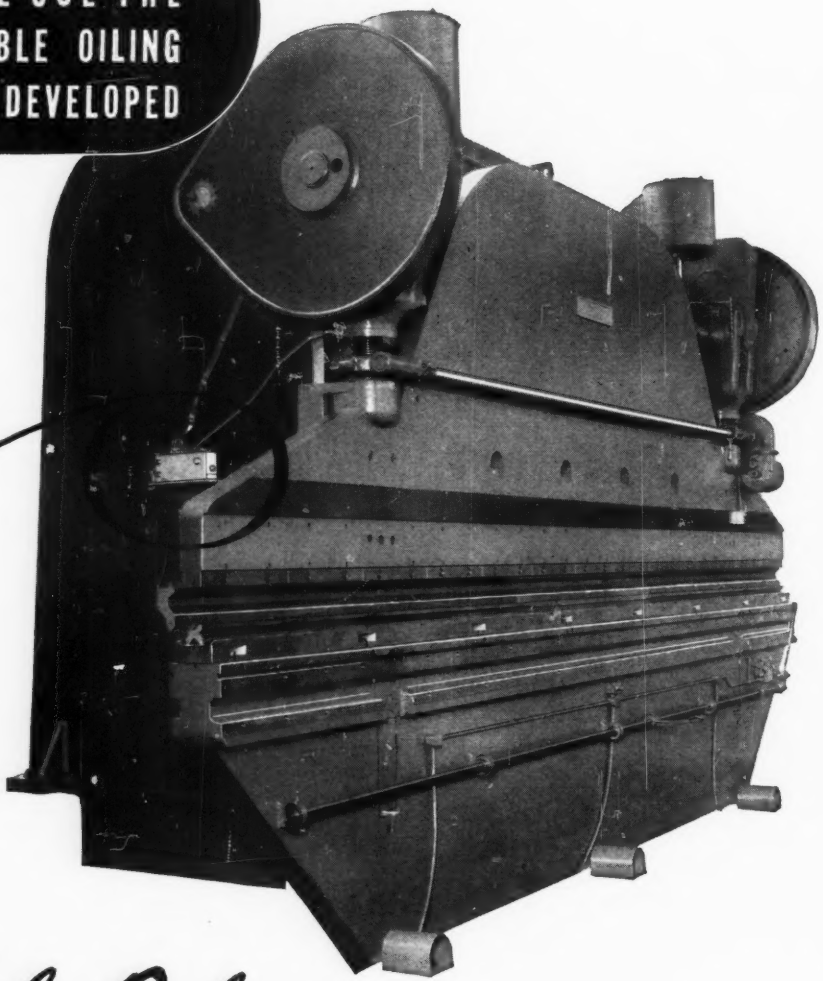

Illustrated is Madison-Kipp Lubricator Model FD installed as original equipment on a 3/8" by 20' Cincinnati Press Brake, manufactured by the Cincinnati Shaper Co., Cincinnati, Ohio.

MADISON-KIPP

Fresh Oil . . . by the measured drop, from a Madison-Kipp Lubricator is the most dependable method of lubrication ever developed. It is applied as original equipment on America's finest machine tools, work engines and compressors. You will definitely increase your production potential for years to come by specifying Madison-Kipp on all new machines you buy where oil under pressure fed drop by drop can be installed.

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the amazing "lubrication fitting" that thinks ahead...

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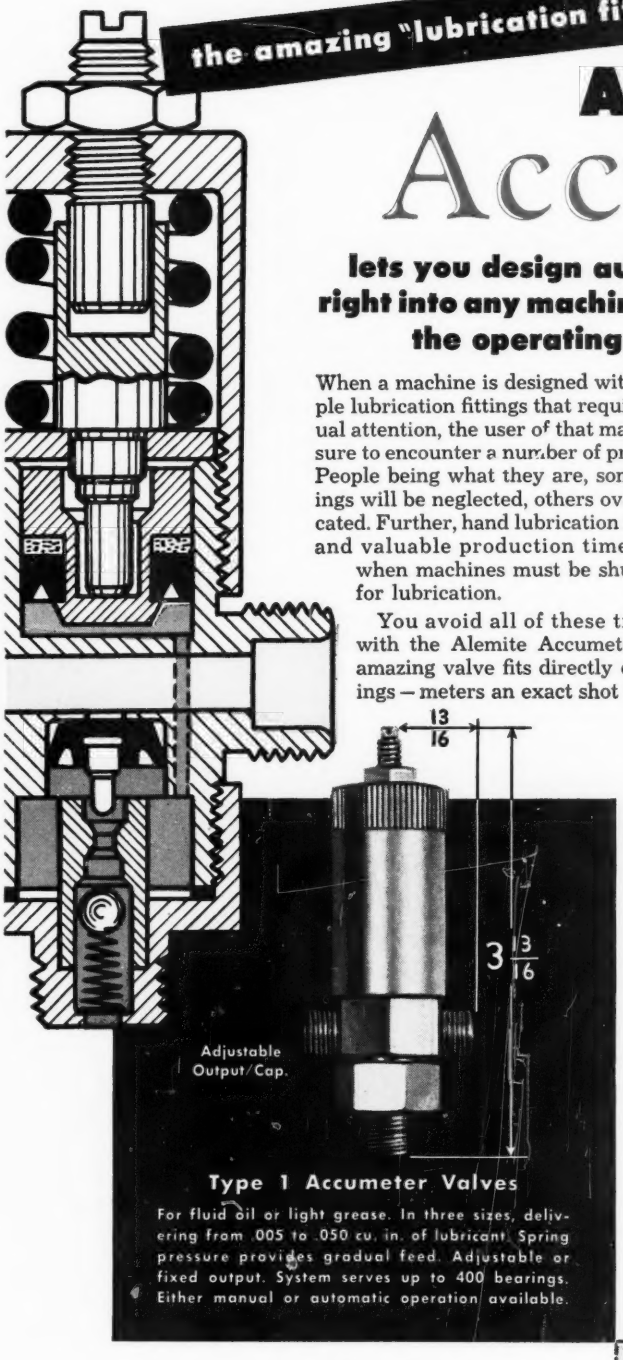
lets you design automatic, fool-proof lubrication right into any machine—simply, economically...offers the operating savings industry will buy!

When a machine is designed with multiple lubrication fittings that require manual attention, the user of that machine is sure to encounter a number of problems. People being what they are, some bearings will be neglected, others over-lubricated. Further, hand lubrication is costly and valuable production time is lost when machines must be shut down for lubrication.

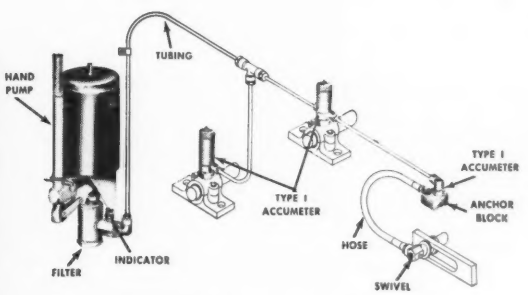
You avoid all of these troubles with the Alemite Accumeter. This amazing valve fits directly on bearings – meters an exact shot of oil or grease *automatically—at pre-determined intervals—while the machine is in operation!* Time, production and maintenance costs are cut to the bone! With all these advantages, it is small wonder that 95% of all major plants buying machine tools specify centralized lubrication.

The Alemite Accumeter system is simple to design and build into any machine. Automatic Accumeter Systems assure you positive, low-cost lubrication. Find out about these automatic systems now. See the savings, the efficiency they add and you too will specify Accumeter!



Type 1 Accumeter Valves

For fluid oil or light grease. In three sizes, delivering from .005 to .050 cu. in. of lubricant. Spring pressure provides gradual feed. Adjustable or fixed output. System serves up to 400 bearings. Either manual or automatic operation available.



offers all these advantages!

- Eliminates shutdown time for lubrication. Adds productive time to machine output
- Seals completely against dirt, grit, water all the way from "Barrel-to-bearing"
- Prevents bearing troubles due to neglect or use of wrong lubricant
- Services all bearings—including those inaccessible or dangerous—in one operation
- Avoids work spoilage and bearing repairs due to over-lubrication

factory tested—field proved

Exhaustive, in-the-field tests show no appreciable variation in the amount of lubricant discharged after 73,312 lubrication cycles—*equal to* 122 YEARS *of twice-a-day service!*

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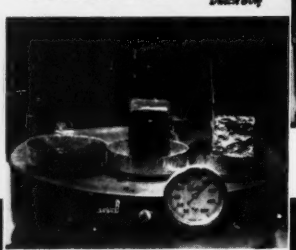
THE picture story shown below demonstrates why multi-purpose Bentone* greases have proved superior to conventional greases in high temperature service. A grease compounded with Bentone 34, a non-soap high-efficiency gelling agent, was heated between two ordinary soap base greases. While the gel structure of the two soap greases broke down as the temperature increased, the thermal stability of the Bentone grease can be seen at even 500°

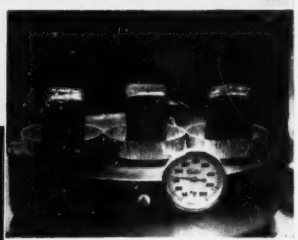
Although the temperature at which Bentone greases will ignite is set by the fire point of the oil (as in the case of conventional greases), Bentone 34 greases, when properly compounded *will not melt*, even while burning.

Write today for your free copy of the new Bentone 34 brochure, filled with complete information about the many advantages of this non-soap gelling agent.

* TRADE MARKS REGISTERED U. S. PATENT OFFICE



The gel structure of conventional greases is breaking down.



Bentone 34 grease, center, between two conventional soap base greases.

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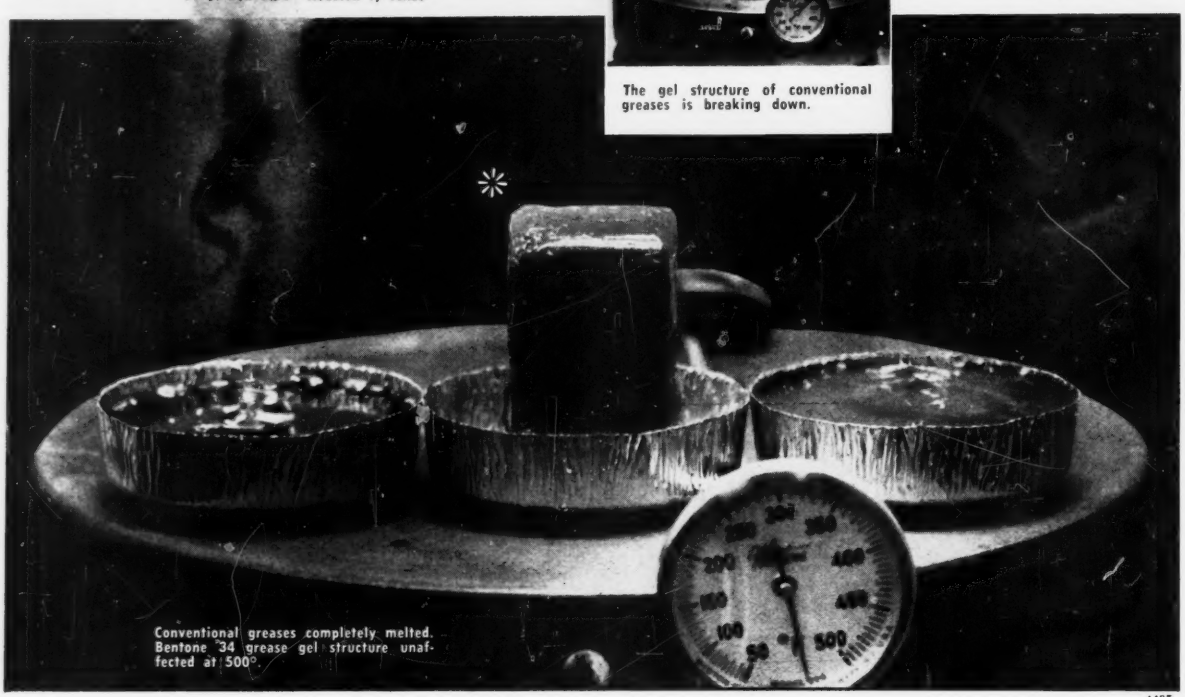
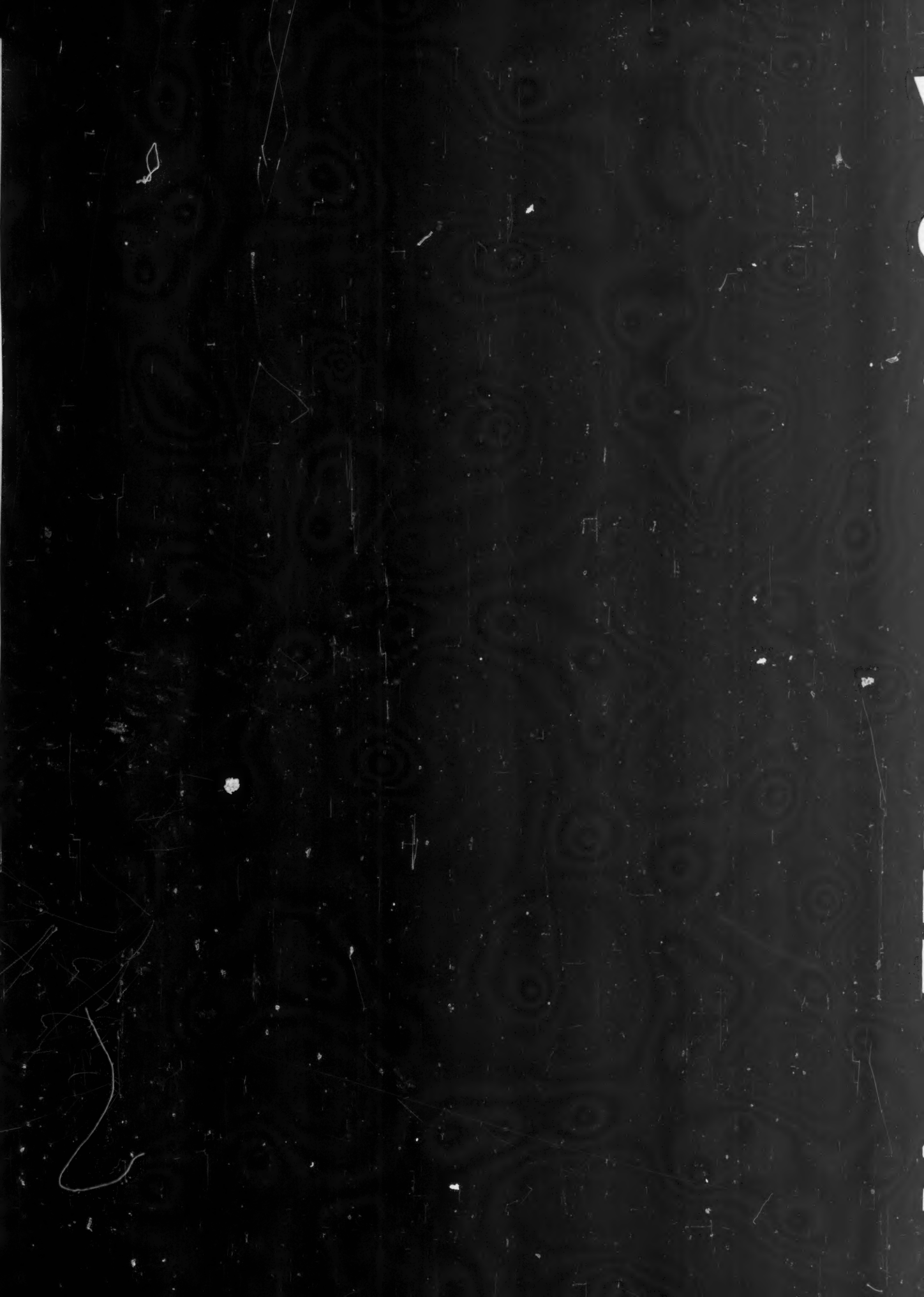
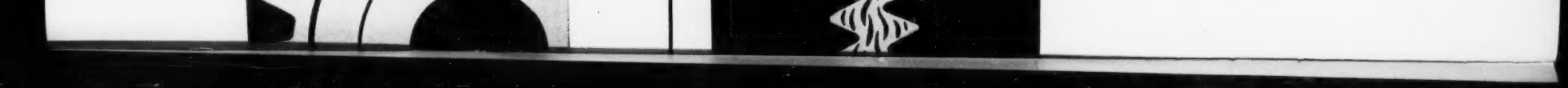
P. O. Box 1675 Houston 1, Texas



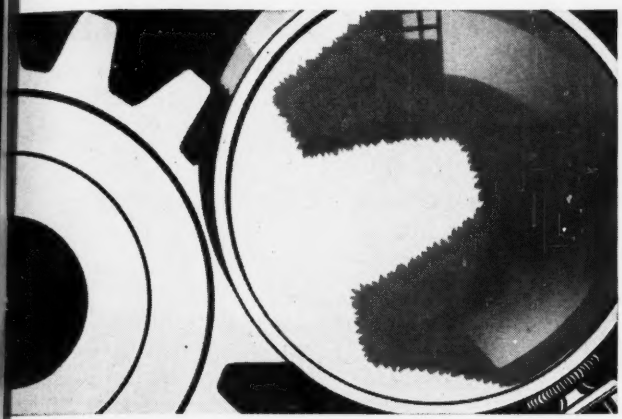
Conventional greases completely melted. Bentone 34 grease gel structure unaffected at 500°.

4495

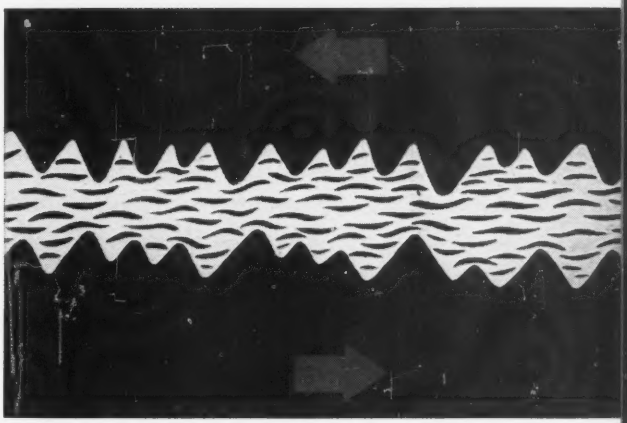




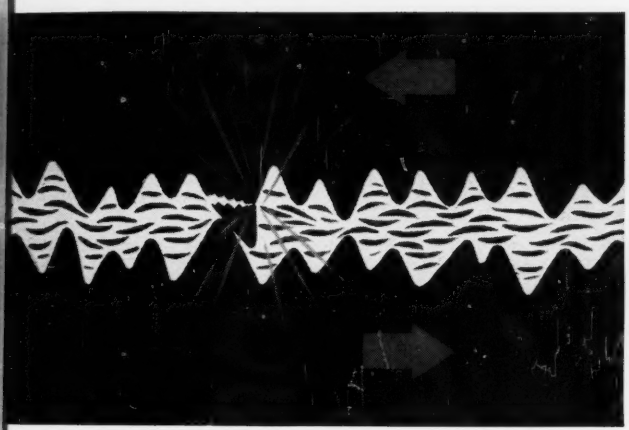
HOW SUNEP PREVENTS GEAR WEAR



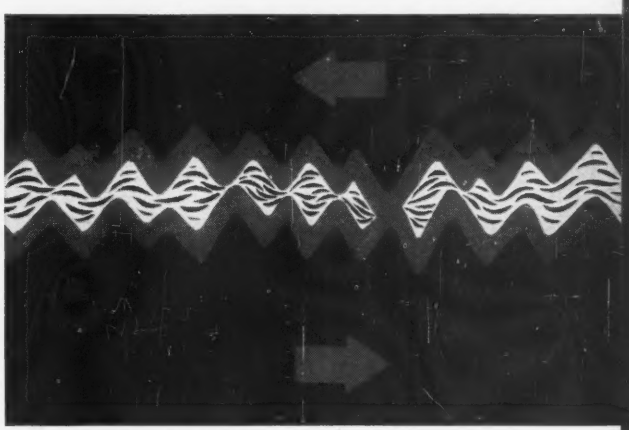
The surface of a gear may seem smooth, but when greatly magnified it looks something like this. The lubricant must keep these jagged edges apart.



With ordinary oil, a thick film is needed to keep these metal surfaces apart. The thickness of the film depends on the viscosity of the oil and other factors.



It is difficult to maintain a thick oil-film in everyday practice. The oil film thins out and semiboundary lubrication results. With ordinary oil, metal to metal contact occurs and gears wear.



Sunep coats the surfaces with a chemically bonded film which is virtually unbreakable. This film, although only about 0.0000001" thick, keeps the surfaces apart, withstands heavy loads and sudden shocks, prevents wear.

Sunep has many advantages. In addition to its outstanding load-carrying ability, it is noncorrosive; it keeps gears and bearings clean; it prevents rusting; it stays put. And its additives do not drop out during use or after prolonged storage. For more information about Sunep and how it protects your equipment, call your nearest Sun office or write SUN OIL COMPANY, Philadelphia 3, Pa., Dept. LE-2.

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1954 ASLE ANNUAL MEETING & EXHIBIT
NETHERLAND PLAZA HOTEL, CINCINNATI, APRIL 5-6-7

A partial list of papers to be presented at the 9th ASLE Annual Meeting & Exhibit include:

- Bearing Design Today
- Behavior Of Sleeve Bearing Materials, by C. M. Allen & K. A. Davis
- Boundary Friction Of Very Well Lubricated Surfaces, by E. Rabinowicz
- Fleet Taxi Engine Wear, by O. Wuerz
- Frictional Properties Of Soap At High Pressures, by J. R. White
- How To Put Across A Technical Paper Or Report, by L. N. Rowley
- Lubrication Application Problems & Their Solution In The Coal Industry (Panel)
- Lubrication Application Problems & Their Solution In The Steel Industry (Panel)
- Metalworking Session, covering wire drawing, wax lubrication & metal-cutting cutting oils
- Micro Testing Of Greases, by H. V. Wadlow & E. E. Campbell
- Railroad Bearing Lubrication
- Reclamation Of Industrial Petroleum Products (Symposium)
- Rust Prevention, by E. J. Schwoegler
- Surface Active Preventives In Lubricating Oil, by D. W. Criddle
- Synthetic Lubricant Clinic

NEW LUBRICATION COURSE TO ACCOMPANY ANNUAL MEETING: The ABC's of Lubrication Engineering, five 3-hour class sessions covering the basic principles of Lubrication Engineering and presented by leading experts in the field, will be offered plant personnel starting Monday afternoon and continuing the morning and afternoon of the following two days. Registrants will be able to attend the regular meeting sessions and exhibits when time permits.

1st Session. Introduction. Motivation of the course. Fundamentals of fluid film lubrication Fundamentals of non-fluid film lubrication. Dry friction. Solid lubricants.

2nd Session. Properties of lubricants. Petroleum oils and greases. Synthetic oils and greases. Tests and their significance.

3rd Session. Application of lubricants. Continuous flow circulating systems. Intermittent flow systems for oils and greases. Commercial types. Lubricating cups and fittings, etc. Maintenance of lubricating systems and lubricators.

4th Session. Bearings and bearing components. Application and maintenance. Ball and roller bearings. Journal bearing grooving. Oil seals, gaskets, and packing. Gear lubrication.

5th Session. Plant lubrication. Lubrication schedules. Storage and handling. Selection of proper lubricants. Analyses of failure. The principles of economic lubrication.

J. J. O'Connor, Chairman

Practical Lubrication, No. 11

TEXTILE MILL LUBRICATION*

by K. P. Powers
Gulf Oil Corp.
Gulf Bldg.
Pittsburgh 30, Pa.

ABSTRACT: Selection of lubricants for textile machinery is reviewed in terms of typical lubricants and their physical properties. The more important applications are discussed in some detail.

The textile industry has learned by experience of the necessity for proper lubrication of the machinery which it operates. Many hours of profitable production have been sacrificed while worn parts have been repaired or replaced because of unsatisfactory lubrication and many dollars have been spent because of excessive friction losses. Today, the need for correct lubrication is recognized to a far greater degree than ever before, and, in general, textile mill lubrication is on a sound basis.

The selection of the right lubricants for a mill is probably more difficult in the textile industry than in any other industry. Reasons for this are: (1) the wide variety of machinery involved, (2) the fact that many machines of ancient vintage are still in use, and (3) the variation of dimensional fit of bearings between machinery builders. The selection is further complicated by the consideration of staining, since lubrication of machinery in the textile field must be accomplished in such a way that the material being manufactured is not contaminated by oil or grease.

Of the various physical and chemical properties that are conventionally used to describe lubricants, viscosity is the most important for oils used in the lubrication of textile machinery. The effect of viscosity is most pronounced in the spinning room where power consumption may amount to as much as 60% of the total power used by a mill. Since conventional spinning frames have 200, or more, spindles, the greatest portion of spinning frame power use is in spindle bearings. Considering this, it is evident that great savings are possible through the selection of the proper spindle lubricants. In order to study spindle lubrication more thoroughly, it is well to examine a number of different types of spindles and the properties of the lubricants recommended for them.

The plain bearing spindle is a type used in many sizes and applications and at speeds up to 11,000 rpm. The Saybolt Universal viscosity (SUS) at 100 F. of oils used in this type of spindle varies from 60 to 220 seconds, depending on kind of service, speed, and size of package. Lubricant No. 1, the conventional characteristics of which are tabulated in Table I, is one used for a plain bearing cotton spinning spindle operating at about 9000 rpm. This type of oil has been jointly selected by the oil refiner, the machinery builder, and the textile industry as being suitable for the spindle on the basis of many years of experience in numerous mills.

The oil has two important functions in this application. First, it must lubricate the spindle with a full fluid film in the bearings but with the lowest possible viscosity, or body, in order that the internal friction of the lubricant, and consequent power losses, may be kept to a minimum. Second, the oil should cushion the action of the bolster in order to absorb vibrations caused by unbalanced bobbin conditions.

An examination of the characteristics of lubricant No. 1 reveals that the only property relating directly to the functions required of the lubricant is that of viscosity. This characteristic has a direct bearing on both lubricating ability and cushioning effect. The other tests are of value to the refiner for control purposes and also indicate to some extent the source of the crude from which the lubricant was derived and the degree and type of refining to which it has been exposed.

Anti-friction bearing spindles of various makes are also frequently found in textile mills. A commonly encountered type is that which is represented by the Whitin Spindle equipped with an S. K. F. Bearing. Spindles of this type are also used in various sizes and speeds. These spindles are lubricated by oils having Saybolt Universal Viscosities at 100 F. of from 60 to 150 seconds. As with plain bearing spindles, the proper selection of lubricant viscosity depends on package and speed. For cotton spinning at about 9000 rpm, lubricant No. 1, Table I, is also suitable for this type of spindle.

The Whitin S. K. F. Spindle has a floating bolster similar to that of the plain bearing type and, therefore, vibration cushioning is also a factor. This is actually the determining condition for the selection of a proper lubricant, since the anti-friction bearing could be adequately lubricated by the least viscous spindle oils commonly available. Remarks made above concerning physical and chemical properties and performance of oils for plain bearing spindles also apply to this type of anti-friction bearing spindle.

The Marquette roller bearing spindle is an anti-friction bearing spindle of another type. The roller bearing under the whirl is positioned solidly in the spindle case and vibration dampening is performed by springs in a floating sleeve type footstep bearing. Since vibration dampening is not a function of the oil in this spindle, the selection of correct lubricant viscosity involves lubrication only. Experience has

*Sponsored by the ASLE Technical Committee on Physical Properties of Lubricants, and presented at the ASLE 8th Annual Meeting, Boston, April 15, 1953.

	LUBRICANT NUMBER											
	1	2	3	4	5	6	7	8	9	10	11	12
Gravity, °API	33.6	34.9	32.5	31.0	29.1	27.0	33.3	21.2	29.6	28.3	22.1	16.3
Viscosity, SUS												
100° F.	73	58	107	80	305	2160.	2050	966	158	610	609	20890
210° F.	36	34.5	39.9	37.6	52.7	143	50.3	72.6	44.5	69.4	59.2	372
Viscosity, CS												
100° F.	13.9	9.8	22.2	15.7	66.0	468	444	209.1	33.9	132.0	131.8	4523
210° F.	2.9	2.5	4.2	3.4	8.1	30.3	7.4	13.7	5.6	12.8	10.0	80.5
Viscosity Index	-	-	97	-	98	100	-	51	113	97	42	-
Flash, OC, °F.	340	315	385	350	450	600	380	395	400	490	390	415
Fire, OC, °F.	380	355	435	400	510	670	410	450	465	560	445	475
Pour, °F.	-5	-5	0	0	+5	0	-25	-5	+10	+5	-15	+40
Color, ASTM Union	<1.0	<1.0	<1.0	2.75	2.0	2.75	Amber	4.0	1.75	2.5	3.5	Black
Carbon Residue, Conradson, %	trace	nil	trace	0.29	0.04	0.45	0.24	0.15	0.05	0.12	0.13	7.12
Neutralization Value, ASTM D 974 Total Acid No.	nil	nil	0.03	1.25	0.07	0.01	0.13	0.05	0.13	0.07	0.05	2.38

Table I Physical and chemical properties of typical textile machine lubricating oils.

shown that low viscosity oils are suitable for this spindle and that only one type of oil is required regardless of speed and size of package, even for heavy duty twister spindles. Lubricant No. 2, Table I, has characteristics suitable for use in this spindle. Of course, a somewhat more viscous lubricant could be used, but an axiom of spindle lubrication is "the lower the viscosity, the less the power loss", so naturally the lowest viscosity spindle lubricant which will do the job is used.

The Universal Winding Company's Uniflex Spindle is an anti-friction bearing spindle of still another type. This spindle also has a roller bearing at the top of the spindle case and a plain bearing footstep bearing. However, the footstep bearing is not free to move. Vibration dampening is accomplished through flexing of the spindle shaft between the two bearings. Lubrication is not provided by the conventional oil bath, but by saturating a felt once a year through a small hole in the seal plate. Since the spindle base is sealed, lubricant loss is very low and it is necessary that the lubricant be stable to oxidation and breakdown under the conditions of operation and, thereby, resistant to the formation of gummy materials. While the manufacturer of this spindle has decided that an oil of approximately 100 SUS at 100 F. is best for this spindle, it is possible that oils of viscosities down to 60 SUS could be used satisfactorily, since no severe lubrication condition has to be overcome. As the bearing does not operate in a bath, internal lubricant friction losses are insignificant. Therefore, the use of the heavier oil results in little power loss. A typical oil for this spindle is lubricant No. 3, Table 1.

During the previous discussion, the selection of lubricants for spindles has been based entirely on viscosity. It should not be assumed that this is the only important oil characteristic to be considered. It is, however, the only one of the conventional physical and chemical properties that has a direct bearing on the application, assuming, of course, that well refined and uninhibited oils only are used. Since relatively small amounts of oil are used for long periods of time in spindle lubrication, oils having good resistance to deterioration pay off in this application.

Recently a new type of oil has been presented for textile spindle lubrication. Properties of this lubricant, which is No. 4 in Table I, are to be compared with lubricant No. 1 since they should be used in similar service. While these two types of oils are alike in several characteristics, the darker color, greater carbon residue, and higher acid number of lubricant No. 4 suggests that it is not as well refined as the other or that it is an additive type oil.

As a matter of fact, this new lubricant is much superior to the other in many respects which are not revealed by conventional tests and it represents a real contribution to the art of textile spindle lubrication. Not only does this improved spindle oil have greater film strength and superior rust prevention characteristics, but it is extremely resistant to deterioration in service and has the unique ability of keeping contaminants from settling out. The end result is spindle operation with a minimum of wear and exceedingly clean spindle bases.

Such variation in performance is entirely unpredictable from the conventional physical and chemical characteristics, and, if we are to evaluate spindle lubricants properly, we must either evolve new methods of laboratory testing or depend on operational experience.

Representative of those lubricants performing in applications where close tolerances exist and where maintenance and replacement costs are important are the comb box lubricants. The friction losses of the comb box are small in comparison with those of the entire carding machine but for good card performance, and proper doffing, it is essential that the comb box be kept in good mechanical condition and that wear be held to a minimum.

The operation and lubrication of a card comb box involves considerable movement of parts, particularly since it operates at speeds of 900 to 1500 rpm. Since so much rapid reciprocating motion is involved, the lubricant must move around freely and quickly reform temporarily ruptured oil films that will adequately lubricate all surfaces. An oil which is too light in body will not give the lubrication protection required and wear will result. One which is too heavy will not only have so much internal friction that excess heat is evolved but also fail to circulate properly through the oil passages and carry off heat from the bearings. Inadequate lubrication and excessive wear will also occur under these conditions.

Lubrication No. 5, Table I, is suitable for card comb box lubrication, but, of all of the properties tabulated in the table, the only one of real significance is viscosity. Again, in this application, stability in service is of importance; unfortunately, none of our conventional tests indicate this characteristic. The oil is used over and over and should be capable of lasting for a long period without de-

	LUBRICANT			
	A	B	C	D
Gravity, °API	31.5	36.6	23.3	14.7
Melting Point, ASTM D 127, °F.	135	-	-	-
Dropping Point, ASTM D 566, °F.	-	190	318	202
Penetration, ASTM Unworked	165	280	-	-
ASTM Worked	-	-	325	312
Viscosity, SUS, 210° F.	57	-	131[1]	64[1]
Flash, OC, °F.	440	320	-	-
Fire, OC, °F.	500	370	-	-
Pour, °F.	125	140+	-	-
Color, ASTM Union	2.5[2]	<1.0[2]	-	-
Carbon Residue, Conradson, %	0.08	0.50	-	-
Neutralization Value, ASTM D 974				
Total Acid No.	nil	5.94	-	-
Type of Soap	-	-	Sodium	Calcium

Table II Physical and chemical properties of grease type textile machine lubricants. Key: [1]Viscosity of oil base, [2]Melted state.

veloping sludge or increasing unduly in viscosity.

A different type of application where replacement and maintenance costs are important is tenter chain lubrication. Not only must proper lubrication be supplied to prevent excessive wear, but the lubricant should be of a type which will not cause the tenter chain to eventually operate incorrectly because of deposits resulting from breakdown of the oil at the high temperatures to which tenter chains are often subjected.

The oil must resist thinning out in the heat evolved and not run off the chain; otherwise, friction losses will be high, wear will take place, and the life of the chain will be unnecessarily reduced. Also, as mentioned previously, the oil must not break down to form deposits which will interfere with the proper operation of the chain. The properties listed for lubricant No. 6, Table I, an extremely satisfactory lubricant for tenter chains, fail to predict completely the performance of the lubricant in the latter respect. Admittedly, the carbon residue is low, for this heavy an oil, but this in itself is not conclusive, since other types of oils with low carbon residues do not have the resistance to sludge formation of the recommended lubricant. Here again conventional tests are of little value in predicting an oil's particular behavior and performance in service is the only way that it can be determined that this is the correct lubricant to use.

Proper lubrication of rolls and saddles of drafting systems is often a complicated affair because of the variety of bearing types that are involved. For the purpose of this discussion, any reference to grease packed anti-friction roll bearings will be omitted as, in general, they are free from the difficulties of leaking. Yarn staining and cot damage are the particular points we wish to stress regarding the lubrication of drafting rolls.

Lubricants No. 7 and No. 8, Table I, are types commonly used for hand oiling drafting rolls. Both lubricants are satisfactory. Lubricant No. 7 is a soap thickened type, while lubricant No. 8 contains an additive to give it adhesive characteristics. Both lubricants have the ability to stay where applied; that is, they do not run off and spread to the rolls where the yarn would be contaminated and where roll coverings would be damaged. This ability to stay where applied cannot be predicted from the typical properties shown and, as a matter of fact, cannot be predicted from any laboratory test of

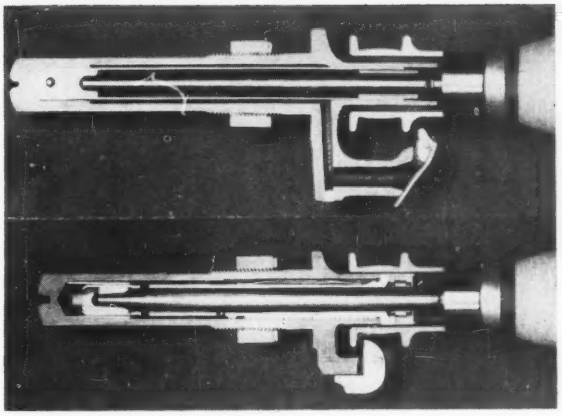

Figure. 1 Textile spindles; (top) Whitin plain bearing spindle, (bottom) Whitin SKF spindle. (Courtesy of the Whitin Machine Works.)

which the writer is aware.

No discussion of textile machinery lubrication would be complete without considering the subject of twister rings. Many rings on spinning and twisting frames are run dry, and under certain conditions, satisfactory results are obtained. On the other hand, in many instances it is necessary to lubricate to prevent undue wear of rings and travelers and to produce a more even yarn with less ends down.

Whitinsville Spinning Ring Company Eadie Rings are lubricated by wicks extending around the rings and supplied by oil from wells. The heavy yarn type has a helical groove into which there is fitted a wick which is reinforced by spring wire to hold it in place. The laced type is for finer yarns, and since the wick in this type of ring comes in contact with the working surface to a lesser degree, the supply of lubricant is naturally less. Lubricant No. 5, whose properties are given in Table I, is generally suitable for both of these rings. It is a high quality mineral oil with inhibitors but not containing compounding in the general sense of the term.

Sometimes it is necessary to increase or decrease the flow of oil to provide correct lubrication of twister rings and this can be done by decreasing or increasing viscosity. That is, the higher the viscosity of the oil, the less the oil flow, wetting characteristics remaining the same. In some cases, oils compounded with materials to increase wettability or oiliness are used, although in general such lubricants are not required.

The Herr Conical Ring is another type of ring lubricated with oil flowing through a wick. However, the oil is supplied at several points only. These points are located flush with the inner and top surfaces of the ring. To provide satisfactory lubrication, a relatively low viscosity product with compounding to provide wettability (increased flow) and oiliness are required. A suitable lubricant is represented by lubricant No. 9 in Table I.

The Albrecht Ring manufactured by Merriman Bros. is a porous metal type of twisting ring in which oil is supplied to the back of the ring by a wick dipping into an oil well and extending around the ring. Two types of oil are used for this ring. Lubricant No. 5, Table I, can be used when twisting continuous filament yarns and lubricant No. 10,

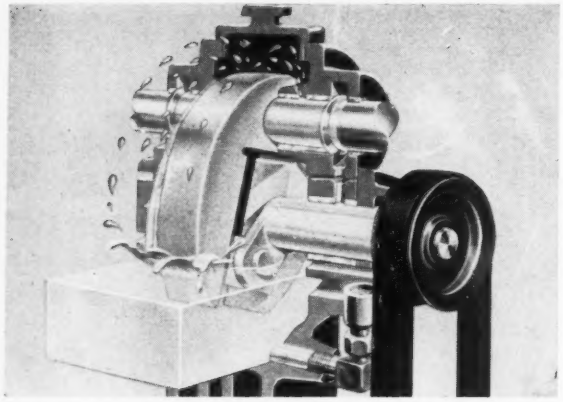

Fig. 2 Textile card comb box. (Courtesy of the Gulf Oil Corp.)

Table I, when twisting staple yarns. These oils are similar in quality; however, the latter is about twice as viscous at 100 F. as the former and is used to combat the tendency of staple yarns to draw off of ring surfaces large amounts of oil, thereby controlling excessive oil consumption.

The Whitinsville Spinning Ring Company manufactures a grease-lubricated ring having multiple grooves to retain the lubricant for a longer period than is possible with a plain ring. Since there are many types of greases on the market suitable for this ring, it would be impossible to show the characteristics of all of them. Petrolatum is often used and in Table II is given the typical physical and chemical properties of a petrolatum (lubricant A). Lubricant B, Table II, is a special wax-oil mixture containing some soap. Both are frequently quite satisfactory for this type of ring.

Occasionally oils are used on grease type rings and their viscosity and compounding may also vary widely. For example, the writer is aware of one plant where compounded steam cylinder oil is used, hand applied to a plain grease type ring. The yarn is rayon tire cord and the doffing period is over eight hours. Many different materials have been tried in this particular application, but this lubricant gives consistently better results than any other type ever used.

Space does not permit a complete discussion of loom lubrication, so the examination in this paper of lubricants for this type of equipment will of necessity be very limited. A typical example of the hand oiled loom is the Crompton & Knowles Cotton Loom equipped with plain bearings. Three different types of lubricants in common use on this loom are lubricant No. 11, Table I, a straight mineral oil; lubricant No. 7, Table I, a soap thickened oil; and lubricant No. 8, Table I, an oil with a special tackiness ingredient. The last two lubricants have come into use because they have the characteristic of staying in the bearing and, therefore, permitting longer intervals between lubrication. When applied in excess, they do not throw off as readily as straight mineral oil, thus minimizing the possibility of fabric staining. Such characteristics, as indicated previously, are not predictable from the typical properties shown.

It is of interest at this point to comment about loom parts equipped with anti-friction bearings with

Fig. 3 Textile spinning and twisting rings; (top) Eadie heavy yarn type, (middle) Eadie laced type, (bottom) Eadie grease type. (Courtesy of the Whitinsville Spinning Ring Co.)

provision for lubrication with a pressure grease gun. The lubrication of such bearings is in general a problem of little consequence and most good anti-friction bearing greases made with reasonably high viscosity base oils are satisfactory. Lubricant C, Table II, is suitable for this application. The use of anti-friction bearings on looms represents a real improvement in loom design with the increased cost easy to justify in reduced maintenance costs. With such an installation the lubrication period is extended and repair costs are minimized.

Two other groups of lubrication points on looms that are worthy of consideration are gears and cams. There are several types of lubricants suitable for this purpose. Lubricant C, Table II, is a conventional sodium soap grease made up with a base oil having an approximate SUS of 131 at 210 F. It is stringy in texture and has an NLGI consistency of No. 1. Lubricant D, Table II, is a calcium soap grease having a base oil with an approximate SUS at 210 F., of 64 seconds. This grease is smooth and buttery in texture and has an NLGI consistency of No. 1. In addition, it is compounded with materials to give it increased load carrying capacity and the ability to resist film rupture under shock load. The third lubricant, oil No. 12, Table I, comes in the classification of straight mineral oils; however, it is high in viscosity (372 Seconds, SUS at 210 F.) and is asphaltic in character. All three of these lubricants have one characteristic in common; they adhere well to the surfaces to be lubricated. Again, such performance would be difficult to predict from the typical properties of these lubricants, which are so widely different.

It is in order at this point to mention that the harder or more solid types of cam and gear lubricants are not always best for looms. When they are used, the housekeeping problem is almost al-

ways accentuated. This may be due to the fact that more of such lubricants can be applied without their running off, and consequently, there is more lubricant to throw off when the loom, with its shock motions, get under way. Another disadvantage of the heavier consistency products is that they are not as able in reforming lubricating films on working surfaces. Best success with gear and cam lubrication of looms has been achieved with soft, adhesive products.

It is concluded from the foregoing that the conventional physical and chemical laboratory tests, with the exception of viscosity, are of little value in selecting lubricants for application in the textile industry. In this industry, perhaps more so than in any other industry, the process of selecting the right lubricant to do a job requires experience in applying lubricants. The oil companies with their knowledge of lubricants and technical contacts with textile equipment manufacturers can be of much assistance to their customers and should be relied upon to recommend proper lubricants as well as to furnish engineering assistance regarding the application of lubricants.

Section News

Northern California November meeting—Dr. W. E. Campbell (ASLE President), Bell Telephone Laboratories, presented a paper entitled The Present Status of Fretting Wear (published in ASTM Special Technical Publication No. 144, 1953). Dr. Campbell (right) is shown discussing his paper with Chairman J. A. Lettier (left), Shell Oil Co., and Program Chairman A. C. West, California Research Corp.

December meeting—Dr. J. L. Dreher (Co-author, R. T. Macdonald), California Research Corp., presented a paper entitled Additives for Lubricating Greases, in which the three principal types of grease additives, rust inhibitors, extreme pressure agents, and oxidation inhibitors were discussed with respect to need, methods of evaluation, and the types of compounds which are effective.

January meeting—S. J. Beaubien (Co-author, Dr. A. G. Cattaneo), Shell Development Co., presented a paper entitled A Study of the Role of the Cutting Fluid in Machining Operations. (Submitted by R. H. Decker, Sec'y-Treas.)

Baltimore November meeting—Tour of inspection of the Westport Station of the Consolidated Gas, Electric Light & Power Co. F. Neat, of Westport Station, pointed out that in the older section the boilers were fed by stokers; that previously lubricated once a day with a calcium base grease, the stoker mechanism is now on a 30-day schedule using a grease containing substantial amounts of carbon black; and that oil purification is provided by means of centrifuges and pressure filters, with considerable differences in oil life found from turbine to turbine. (Submitted by J. O. McLean, Vice-Chrmn.)

Chicago October meeting — A "Stump the Experts" session with Panelists C. Garney, Socony-Vacuum Oil Co.; R. A. Kraus, Republic Steel Co.; J. E. Rosnell, Texas Co.; and H. Curtis, D. A. Stuart Oil Co., Ltd. J. W. Peterson, Standard Oil Co. (Ind.), C. C. Blaisdell, Penola Inc., and P. V. Toffoli, Pure Oil Co., served as judges.

November meeting — J. M. Kuntz, Standard Oil Co. (Ind.), presented a paper on the Economics of Petroleum, an analysis of the factors which enter into the market prices of petroleum products, noting the necessity for supplying the consumer with adequate technical advice on products he purchases, and indicating how this affects the total cost of the marketed item. (Submitted by W. P. Green, Sec'y-Treas.)

Cincinnati November meeting — A. S. Morrow, Shell Oil Co., presented a paper entitled Cleaning & Maintenance of Hydraulic Systems, in which he stated Shell has made surveys among their customers relative to preventive maintenance programs on hydraulic systems and found few programs in operation; that management is more concerned about production than maintenance. The importance of thoroughly flushing and cleaning the hydraulic system before the initial charge, as well as keeping the hydraulic oil clean in the system by the use of proper filtering equipment, was stressed. (Submitted by H. F. Yoder.)

Fort Wayne December meeting—A. Sonntag, Sonntag Scientific Corp. & Alpha Corp., presented a paper entitled Interesting Properties & Uses of Molybdenum Disulphide for E. P. Lubrication, followed by a tour through the Centlivre Brewing Corp. (Submitted by J. W. Sites, Chrmn.)

Kansas City November meeting—A "Stump the Experts" session with R. R. Matthews, Independent Oil Compounders Assoc., as Moderator, and Panelists C. J. Boner, Battenfeld Grease & Oil Corp.; A. F. Lyster, Socony-Vacuum Oil Co.; and L. M. Reiff, Bowser, Inc.

Traveling members of ASLE are welcome and invited to attend meetings held the third Monday evening of each month at Benish' Twin Oaks, 5050 Oak St. (Submitted by L. M. Reiff, Sec'y-Treas.)

Kingsport December meeting — R. E. Knopf, Dow Corning Corp., presented a paper entitled Silicone Lubricants. (Submitted by J. E. Fleenor, Sec'y-Treas.)

Milwaukee December meeting—Annual Christmas Party. (Submitted by R. W. Schroeder, Sec'y.)

New York October meeting—A. M. Southcott, Chairman, announced plans for a local educational program on Lubrication Engineering to begin in February

(Continued on page 40)

SOME LUBRICATION PROBLEMS WITH ROLLER BEARINGS AND THEIR SOLUTION*

by R. H. DuBois
Bower Roller Bearing Co.
3040 Hart Ave.
Detroit, Mich.

When one attempts to show typical examples of bearing failures which can be attributed to improper lubrication, certain handicaps are encountered. Failures due to faulty lubrication frequently resemble those due to poor adjustment, misalignment, overloading and other application hazards. This is especially true with respect to tapered roller bearings which will be discussed in this paper.

Fig. 1 shows a typical tapered roller bearing. As such a bearing acts as a fairly efficient centrifugal pump, it is desirable and necessary to provide a return for the lubricant, so the bearing does not starve itself by the pumping action. Troubles emanating from the neglect of this consideration are design shortcomings, not actual lubrication failures.

Passenger cars and trucks use great numbers of tapered roller bearings as wheel bearings. Such an application is shown in Fig. 2. Almost without exception these are grease lubricated bearings. Bearing manufacturers usually prefer to see a light grease of the order of NLGI No. 2 (worked penetration 265-295) used in most applications. Most vehicle manufacturers concur in this preference and apply such a grease in new units. However, in the case of large trucks, heavier greases are used in some fleet operations. This is generally because of inadequate seals. With high temperatures resulting from sustained high speed operation, light greases become very fluid and have a tendency to leak through the seals. With this condition existing, operators turn to a heavier grease. The heavier grease provides a better seal and under higher temperatures has less inclination to become fluid. Where conditions, involve dust and dirt, plus moisture, there is the tendency to turn to an even heavier grease. The heavier grease tends to channel and provide a minimum of oiliness to the bearing. It is a common complaint from heavy truck operators that when a wheel bearing is disassembled for inspection, a complete lack of oiliness is evidenced on the bearing surface of the outer rice. If a more fluid grease were used, it is felt that this complaint could be eliminated.

In connection with the wheel bearings, just mentioned, an epidemic developed several years ago. Certain heavy truck wheel bearings were failing at 12,000 to 100,000 miles. An examination revealed that prior to failure, a condition such as shown in Fig. 3 existed. The stain or etch marks are clearly visible. The subsequent failure or spalling occurred at those points where the stains occurred. The spalled area exactly coincided with the spacing of the rollers, as did the etched marks. Research proved that the seals were allowing moisture to enter the bearing cavities as the bearing cooled and the etching followed.

Regarding the visual detection of failures or impending failures on wheel bearings, this is a fairly easy job if a preventative maintenance program is in effect. Some of the bigger fleet operators inspect wheel bearings every 20,000-30,000 miles. At this time, spalled or fatigued areas can easily be detected on the rollers and outer race, see Fig. 4. The spalled regions are characterized by actual removal of metal. In such cases, the bearing should unquestionably be replaced. If stain marks or etched areas are observed, the bearings should also be replaced and a study made of the seal and the lubricant. A general discoloration of the bearing surface is a definite red flag. Bearings have been observed which are blued, due to heat, but have not yet failed. If there is any doubt as to whether the discoloration is stain or temper, this may be dispelled by washing in commercial solvent. The solvent will remove the stain but of course will not affect the temper color.

Axle and transmission bearings are generally supplied with an ample quantity of lubricant. However, design engineers are continually confronted with the problem of lubricant flow to the bearings at certain speeds and various attitudes. With some of the high preloads now being used on pinion bearings, the lubrication is especially important. If the front pinion bearing is not properly lubricated, it can literally burn up in a very short time. Fig. 5 shows a typical tapered roller bearing equipped rear axle. It is easily seen how far the front bearing is from the source of lubricant, which is in the banjo. Some means of trapping oil thrown up by the ring gear must be contrived and this trapped oil directed to the front bearing.

In axle or transmission applications, visual indications are ordinarily not employed. In passenger cars or trucks, no attention is usually paid until a breakdown has occurred. Noise is the most readily noticeable evidence of bearing difficulties. However, sound telegraphs so well in most vehicles, it is exceedingly difficult to isolate the origin of the noise. Axle bearings in trouble will often whine or squeal loudly enough to be heard above the other usual noises.

Although it is not necessarily a visual means of detecting trouble, dirt or grit in an axle or transmission lubricant can usually be expected to cause trouble. If the lubricant contains grit or foreign

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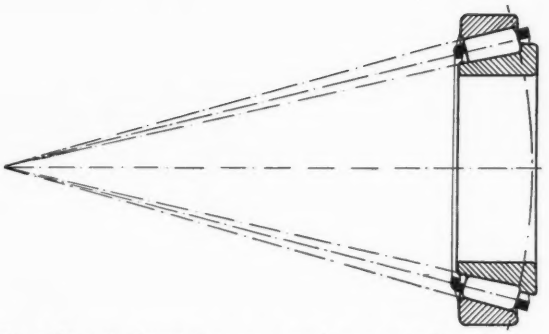


Fig. 1 Sketch showing geometry of tapered roller bearing.

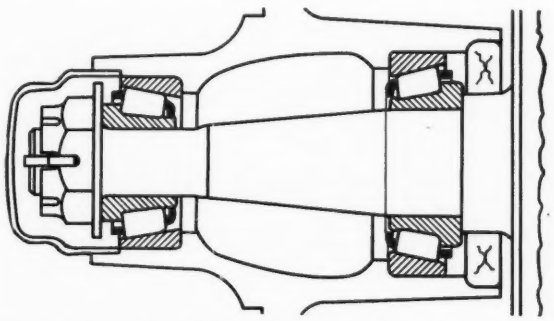


Fig. 2 Automotive front wheel hub and spindle showing application of tapered roller bearings.

matter, the bearings may wear very rapidly disturbing the original adjustment and generally causing trouble. This is especially true of axle bearings which are preloaded.

Overheating is a good indication of bearing illness. Wheel, axle, and some transmission bearings which are in trouble can often be detected by their temperature. However, one must remember that 120 F. is uncomfortable to the hand but does not necessarily indicate an abnormally high bearing temperature. Some truck axles operate at temperatures as high as 200-225 F. in normal operation.

Considerable knowledge of E.P. lubricants has been gained as a result of many C.R.C. hi-torque, low speed tests which have been conducted. It will suffice to say that, if a lubricant is a good gear lubricant, it will also be a good bearing lubricant. Foaming is undesirable. Corrosion is taboo. If wears occurs on gears, tests show that bearings will also wear.

One of the most interesting bearing applications is that of the cylindrical roller bearing as applied to aircraft gas turbines. Operating requirements have forced the development of new steels and new cage materials. It is probable that the next few years will see some interesting design innovations. D-N values (bore in millimeters x rpm), close to two million, temperatures of 600 F., rubbing velocities of 9,000 ft. per minute, loads of very small magnitude, all combine to give roller bearing manufacturers one of their most difficult assignments.

The development of steels, cage materials, and new designs gives no advantage unless there is a lubricant which will permit operation with these innovations. Right now, lubricants are needed which will allow 700 F. operating temperature. The bearing companies are prepared to furnish bearings

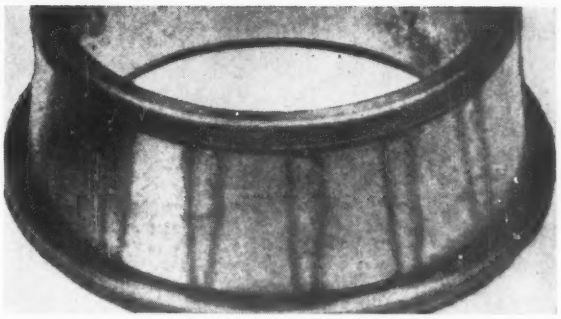


Fig. 3 Tapered roller bearing cone etched by moisture or acid formation.

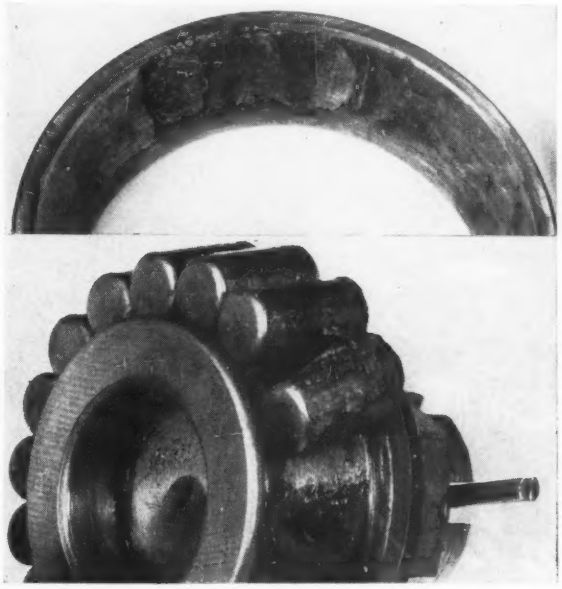


Fig. 4 Spalled or fatigued areas on rollers and races of tapered roller bearing.

which will stand this temperature if they can be lubricated. In addition to the development of such a lubricant, more work needs to be done by bearing manufacturers, engine builders, and oil companies, to devise better means of getting lubricants to the critical surfaces.

These are problems at hand. They have a lot to do with the failures which are now to be discussed.

One of the biggest difficulties associated with high speed roller bearings is that of cage or retainer guidance. There are three methods, namely: roller guided, inner race guided, and outer race guided.

When the author's company in conjunction with gas turbine engineers started to develop a high speed roller bearing, a super precision bearing was used but it was equipped with the old standby retainer material, a leaded brass known as D.K. brass (the D.K. refers to door knob). As early as 1945 a member of the Rolls Royce organization warned that difficulty would be experienced when operating temperatures exceeded 325°. At that time, work was started on new cage materials. As bearings were produced and run in engines, it was soon verified that standard cage materials were limited as to

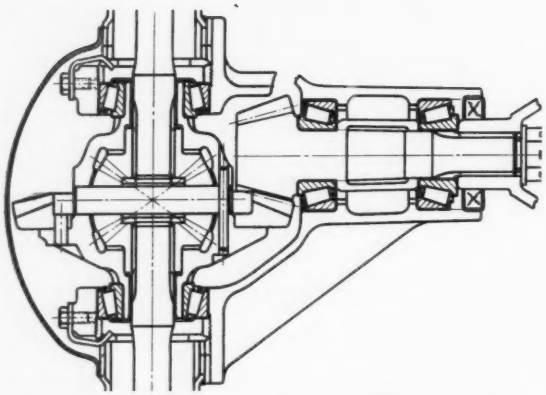


Fig. 5 Automotive rear axle showing application of tapered roller bearings.

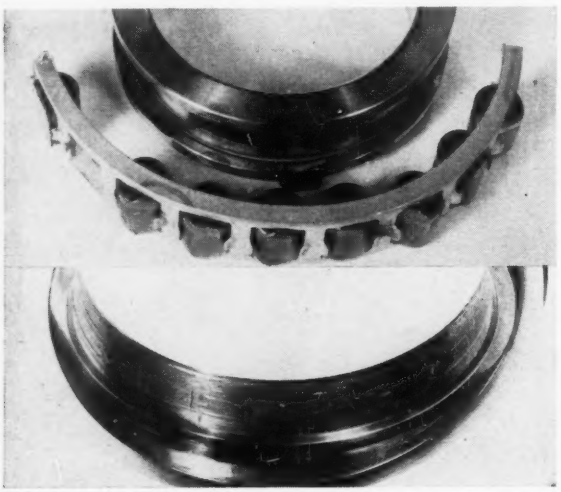


Fig. 6 Smearing of rollers and races of cylindrical roller bearing due to seizure of cage.

temperature. Failures began to occur. Some could be caught in the very early stages and it was noticed that there was considerable transfer of brass to the steel, cage-guiding surface. It was believed that lead separation at quite low temperatures preceded the actual brass pickup. This condition would result in cage seizure and possibly breakage, and a wreck, such as Fig. 6, would occur. Now if lubricant could be directed to the guiding flange to remove the heat, it might have been possible to get by. But the engine builders had certain limitations as to quantity of oil available and the manner in which it could be applied. In most every case, oil was supplied on the "cool" side, or the side away from the the turbine wheeel. Due to high speeds and resultant windage, it was improbable that the opposite, or "hot", side of the bearing received much in the way of needed lubrication. This was amply verified by examination of some of the distressed bearings. In some bearings it was found that one end of the roller showed considerable wear while the other seemed to have barely contacted the inner race flange, Fig. 7. Cages invariably showed the most wear or brass pickup on the "hot" side. It is expected that present laboratory tests will prove that oiling from both sides is important and a necessary step

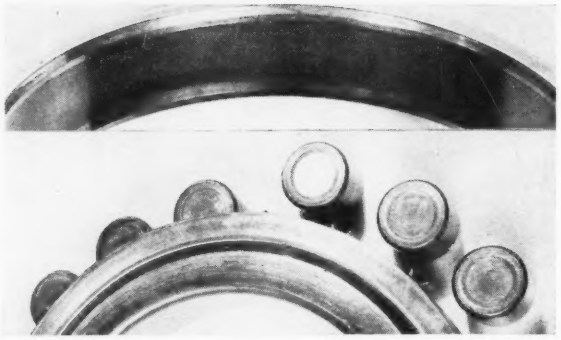


Fig. 7 End wear of rollers due to insufficient lubrication.

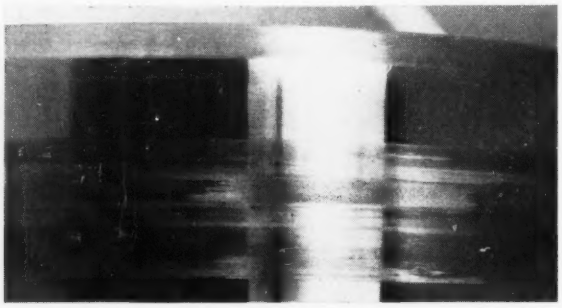


Fig. 8 Circumferential smear marks on race of cylindrical roller bearing.

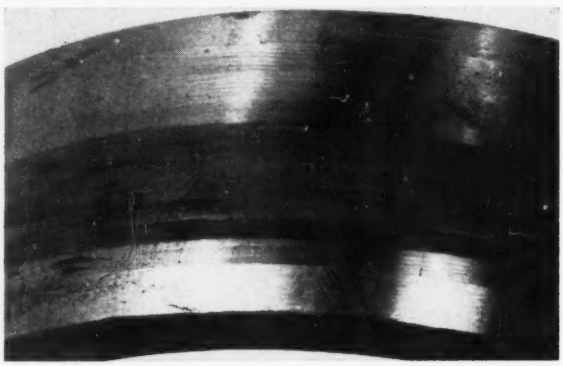


Fig. 9 Circumferential smear marks on race of cylindrical roller bearing.

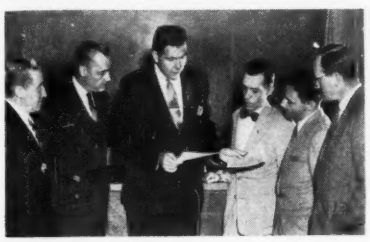
in the successful operation of a high speed roller bearing.

Figs. 8 and 9 illustrate a common and puzzling problem which confronts bearing manufacturers. It is called "frosting", "smearing" and a number of more profane names. It occurs in engines and test stands and is associated with high speed and a lack of radial load. One explanation of this effect maintains that it is simply fatigue, probably due to indeterminate stresses. These stresses could be imposed by manufacturing variations in the bearing; such as, roller diameter, length, track width, and taper cage tolerances; shaft unbalance and all the other possible variations. However, it is believed that if these surfaces were properly lubricated, the "smearing" or "frosting" might not take place. In order to prove this a lubricant is needed which will stick to the bearing surfaces. Some of the deficiencies due to lubrication or materials have been tem-

porarily alleviated by plating. At the present time silver plating and lead plating over the silver are being used, and it is difficult to predict what the third layer will be. It is felt that steels, cage materials, and lubrication should be developed to preclude the necessity of plating.

The visual means of detecting distress in a high speed roller bearing are good. The transfer of metal, brass to steel, or steel to steel is a good indication of trouble. Unusual wear on any surface is a red flag. Discoloration, especially bluing, is a danger signal. Scuffing, galling, or frosting may also indicate failure is close at hand. With the overhaul periods at the present frequent intervals in aircraft gas turbines, it is probable that most bearings which are in trouble may be caught before failure occurs. However, it must be remembered that these bearings, turning at 7,000-10,000 rpm, can develop trouble and fail in a very short time. A means of predicting these failures has not at this time been developed. Upper limits for operation of high speed roller bearings have been set. These limits are not necessarily the maximum, but represent indications of approaching a maximum. Rubbing velocity of the roller in the cage pocket should not exceed 9500 feet per minute. With existing lubricants and design, it is believed that excessive wear of the cage will result if rubbing velocity exceeds this figure.

Personals



D. D. Fuller, Columbia University, has been appointed to the staff of the Mechanical Engineering Division of the Laboratories of The Franklin Institute, where as Principal Scientist he will be in charge of the Friction & Lubrication Section; this is in addition to his professorship at Columbia. Under the auspices of ASLE and The Franklin Institute Laboratories, Mr. Fuller presented a course in the Fall of '53 on Advanced Lubrication; above picture from this session, Fuller third from left.

R. R. Slaymaker, Professor of machine design at the Case School of Applied Sciences, and Consultant for the Cleveland Graphite Bronze Co., has been awarded a machine tool design and economic value award by The National Machine Tool Builders' Association for his paper entitled Bearing Design Using Concentric Journal Theory.

Acheson Industries, Inc., has announced the re-election of **H. A. Acheson** as President and Chairman of the Board, and the appointments of **R. Szymanowitz, P. C. Buck,** and **M. W. Reynolds** as Vice-Presidents; **J. C. Sprague** as Secretary-Treasurer; and **Mrs. L. Tuer** and **L. H. Miller** as Assistant Secretary and Assistant Treasurer.

W. W. Gary has been elected President and Chief Executive Officer of the Attapulgus Minerals & Chemicals Corp., with offices at 210 W. Washington Sq., Philadelphia.

Honan-Crane Corp., Lebanon, Ind., announces the following Sales Engineering staff appointments: **B. J. Hill,** to cover Minnesota, North & South Dakota, Iowa, western Wisconsin, and the Tri-Cities area, with headquarters in Minneapolis; **J. L. Woll,** covering western Pennsylvania and the Wheeling area, with headquarters in Pittsburgh; and **L. A. Gerber,** covering southwestern Ohio, southeastern Indiana and eastern Kentucky, with headquarters in Dayton.

S. Messer, of Oil City, Pa., has been elected Chairman of the Board for Quaker State Oil Refining Corp.

Acheson Colloids Co., Port Huron, Mich., has announced the following appointments: **B. C. Dickey** as Process Development Engineer responsible for improving and developing process operations for current and new products; and **F. M. Hunter** as Service Engineer servicing all of northern and lower Michigan, with headquarters at 3000 E. Grand Blvd., Detroit.

F. D. Koons, District Manager of the Gulf Oil Corp. has retired after 35 years of service.

Obituaries: **H. S. Whitcomb,** Alemite Co. of Ohio Valley, Fall '53.

L. G. Benton, formerly of E. I. duPont deNemours, Inc., and an Honorary Member of ASLE, January 12.

Our new address:
84 E. Randolph St.
Chicago 1, Illinois

Book Reviews

The Atomization of Liquid Fuels

by E. Giffen & A. Muraszew; John Wiley & Sons, Publishers, 440 Fourth Ave., New York 16, N. Y., 1953, 246 pages, price $6.00.

The Atomization of Liquid Fuels gives a concise survey of the characteristics, properties, and most useful results of the atomization of liquid fuels. The authors discuss the mechanism of disintegration of liquid jets, fuel spray characteristics, effect of atomizer on spray characteristics, simple theory of the swirl atomizer, and the use of dimensional analysis for correlation of atomization data. Other topics covered are the effect of physical properties of the liquid on spray characteristics, effect of the properties of the gaseous medium on spray characteristics, effect of injection pressure on spray characteristics, formation and development of intermittent and continuous sprays, and experimental methods for assessment of fuel spray characteristics.

Symposium on Fretting Corrosion (Aug. 1953)

by T. E. DeVilliers, W. E. Campbell, J. R. McDowell, O. J. Horger, E. W. Herbek, Jr., & H. H. Uhlig; American Society for Testing Materials, Publishers, 1916 Race St., Philadelphia 3, Pa., 1953, 88 pages, price $2.75.

Fretting corrosion may be very generally defined as a surface damage that occurs when two solid surfaces are contacted under relatively high loads and with small relative (vibratory) motions. The phenomenon has been recognized by investigators for many years, but even today it is not thoroughly understood. Fretting corrosion has also been called "friction oxidation," "wear oxidation," "false brinelling," "chaffing," "bleeding," "cocoa." Some difference of opinion exists as to the role of corrosion or oxidation in the mechanism of fretting corrosion; some investigators hold that corrosion or oxidation is an integral part of the mechanism, while others believe that it is only incidental.

(Continued on page 39)

A TOOL-BLADE WEAR TEST FOR CUTTING FLUIDS*

by L. C. Hays, Thompson Prods., Inc.
2196 Clarkwood Rd., Cleveland 3, O.
E. J. R. Hudec, Case Inst. of Tech.
10900 Euclid Ave., Cleveland 6, O.

ABSTRACT: A method of evaluating cutting fluids on a lathe is presented. It involves measuring the wear of a blunt tool blade which is rubbed against a freshly cut surface under an applied load. In contrast to conventional cutting adequate flow of cutting fluid to the test surface is assured and the wear on the tool blade is believed to be a measure of the lubricating and cooling ability of the cutting fluid.

The two major functions of a cutting fluid in metal cutting operations are to lubricate the tool and to carry away excessive heat. As a result of this, production rate can be increased, longer tool life, better finishes, closer tolerances, and lower tool and production costs are obtained.

The purpose of this investigation[1] was to devise an objective method to evaluate the lubricating effect and the cooling effect of a cutting fluid. Tool life tests evaluate the net effect of a cutting fluid. How much of the net effect is due to cooling and how much is due to lubrication is unknown. Two different cutting fluids having about the same net effect may each be superior in different characteristics, whereas if the best features were combined, a superior cutting fluid may be developed.

With conventional methods of cutting fluid application, it is uncertain whether the cutting fluid is getting into the rubbing zone between the work and tool. In this proposed test the cutting fluids are being evaluated under rubbing conditions approximating those in the rubbing zone. This method avoids the uncertainty of getting the cutting fluid into the cutting zone with conventional application methods and insures evaluation of the cutting fluid under a condition where it is known that the fluid is getting between the work and tool.

The method employed in these tests is not known to have been presented by any other investigators. Tingle[2], to evaluate lubricants, used a loaded hemispherical slider following a tool cutting a shallow groove. The conventional lubrication testing machines, such as the Almen, SAE, Timken, Falex, and Four Ball, employ standardized, hardened and ground steel rubbing components which are used to measure a lubricant's film strength. The M.S.E. Schlesinger Machinability Tester[3] uses a 10 m.m. diameter ball, rubbing against a work piece on a lathe under 40 pounds pressure for a period of 20 seconds at 400 FPM, and gives a measure of the abrasive properties of a work material.

In the test method being presented, a tool-blade, made of any selected tool bit material is mounted on a pivoted arm such that any desired load can be placed upon it. The tool-blade rubs against a revolving work piece that is wet with cutting oil. The whole arm assembly is mounted on the carriage of the lathe so that the tool-blade can be made to follow a cutting tool. In this way the tool blade is always rubbing on a freshly cut metal surface. By this means, it is felt the tool-blade encounters approximately the same rubbing conditions as a tool bit when cutting metal except that adequate lubrication is assured in the test. By observing the rate of wear and the amount of wear on the tool-blade under a fixed set of cutting conditions, a measure is made of the cutting fluid's cooling ability. A schematic of the test setup is shown in Fig. 1.

The main purpose of this paper is to present this method of evaluating cutting fluids. The results are not comprehensive, in that only one base oil with two different percentages of additive were tested, and the tests were run at only one condition of speed and feed. Obviously much investigation remains to be done on different cutting fluids, different tool and work materials, and different cutting conditions. It is hoped that this presentation will stimulate interest and discussion in this method.

Description of Apparatus. These tests were conducted on a 20" x 67" Axelson Lathe driven by a variable speed D.C. motor. A base plate mounted on the saddle of this lathe formed the support for the bearing pivot block. A pivot arm was mounted on the bearing shaft with the forward end of the pivot arm machined to hold an adjustable holder for the tool-blades. The holder was located so that the tool-blades were tangent to the work at the midpoint of the blades. MoMax H.S.S. cutoff blades used in the test were ground to a width equal to twice the feed used on the tests. An aluminum arm was attached to the pivot arm to carry the load tank. A solenoid valve and globe valve were used to control the water flow into the load tank. A drum filled with water was suspended overhead to insure a constant head of water to the globe valve. This arrangement gave a continuous uniform loading rate of 10 pounds per minute.

Mounted on the pivot arm was a separate arm connected to the pneumatic transmitter unit for measurement of tool-blade displacement. This separte arm eliminated measuring errors due to bending of the load arm. The air line supply was brought through a regulating valve preset at 15 pounds pressure to the transmitter, and from there to the Brown Pneumatic Recorder.

To be certain the surface was just wet with oil when making test runs for lubrication only, cutting oil was dripped on the surface of the work, and the excess removed by a cork wiper rubbing against the

*Sponsored by the ASLE Technical Committee on Fluids for Metal Working, and presented at the ASLE 8th Annual Meeting Boston, April 15, 1953.

a. Cutting Tool Material - MoMax H.S.S.
b. Tool-Blade Material - MoMax H.S.S.
c. Blade Width - .062"
d. Blade Width at Wear Zone - .030"
e. Feed - 0.15 IPR
f. Tare Weight of Arm - 15 lbs.
g. Loading Rate on Arm - 10 lbs. per min.
h. Lever Arm Ratio - 10 to 1
i. Cutting Speed - 100 FPM
j. Work Material - 52100 spheriodized anneal at 190 Bhn.
k. Depth of Cut - .010"
l. Tool Bit Angles - B.R. 10°, S.R. 11°, F.C. 6°, S.C. 6° ECEA 10°, SCEA 10°, Nose Radius 3/64"
m. Oil Application - 1. Drip can
2. Two oil jets delivering oil at a rate of 0.167 Gals. per min. through orifices 0.100" by 0.010"

Table I Summary of Test Conditions.

1. Cutting Oil A
a. Sulfur - .10%
b. Chlorine - 0%
c. Essentially neutral
d. Saponification number - nil
e. Viscosity
1. 100° F. - 104.1 SUS
2. 210° F. - 39.7 SUS

2. Cutting Oil B
a. Sulfur - 2.4%
b. Chlorine - 2.0%
c. Same as Oil A
d. " " " "
e. " " " "

3. Cutting Oil C
a. Sulfur - 4.8%
b. Chlorine - 4.0%
c. Same as Oil A
d. " " " "
e. " " " "

Table II Summary of Cutting Oil Specifications.

work. On tests where oil cooling was used, additional oil was supplied at a rate of 0.167 GPM from a cutting oil tank, under 30 psi. air pressure through two small tubes impinging cutting oil on each side of the blade at the point of contact between work and tool-blade. The ends of the tubes were flattened to give a wide stream.

Test Procedure. Each tool-blade was mounted in the holder with the arm in a horizontal position, and the tool-blade was clamped with the wear zone pressed firmly against the work. A cutting-tool making a light cut immediately ahead of the tool blade insured a nascent surface for the blade to rub against.

At the start of each test, a small cut was made, the tool-blade clamped in position, the spindle was started, the feed engaged, and, at the instant the tool-blade was allowed to rub against the work, a switch was closed which started the timer, and also opened the solenoid valve which allowed water to flow into the loading tank. The test was stopped when the chart on the Brown Indicator indicated the blade was worn to a point where the wear arc approached the width of the tool-blade. The tool-blade was then removed from the holder and after cleaning was reweighed on an analytical balance to determine the total weight lost during the test.

Discussion of Results. A good indication of how reliable any test results are, is shown by how well the results can be repeated. In most experimental work, results cannot be repeated exactly because of the action of a number of minute variables that affect the test conditions.

Fig. 2 shows a typical set of four repeat runs made with cutting oil A. The ordinate is in units of displacement on the circular chart which is 0.00083 in. per unit. At two minutes on the abscissa, which is near the ultimate on this test, the average of the four runs is 27.5 units, with the highest being 30 units and the lowest 25 units. This is a variation of 9% from the average. Since many experimenters consider a variation within 10% as being good, this indicates that this test has good repeatability.

Fig. 3 shows the results of tool-blade wear tests when using three cutting oils, designated A, B and C. When the tool-blade was lubricated only with oil A, the tool-blade wore away at a much faster rate than when it was cooled by the twin jets in addition. After 2¼ minutes, the lube-and-cool curve showed 47% less wear than the lube-only curve. For the same amount of wear, say 10 milligrams, the lube-and-cool ran 29% longer than the lube-only curve. With cutting oil B, it can be seen that both tests, lube-only and lube-and-cool, ran for a longer time than cutting oil A. After 3¾ minutes with cutting oil B, the amount of wear of the lube-and-cool test is 56% less than the lube-only curve. For the same amount of wear, say 10 milligrams, the lube-and-cool curve ran 33% longer than lube alone. With cutting oil C, the lube-and-cool curve has 36% less wear as compared to the lube-only curve after 4½ minutes, and for the same wear, say 10 milligrams, the lube-and-cool curve ran 21% longer than lube-only.

From Fig. 3, the more additive used, the slower the rate of wear on the tests with lubrication only. At 25 milligrams of wear, the 10% additive oil B ran 66% longer than the oil A with no additive, and the oil C with 20% additive ran 146% longer than the oil with no additive. In general terms, we can say this is only a 70% reduction in rate of wear for each of the two 10% additions of sulpho-clorinated additive.

On the cooling and lubrication tests, again the more additive in an oil, the slower the rate of wear. Under this condition, again taking the time values at 25 milligrams of wear, the 10% additive oil ran 75% longer than the oil with no additive, and the 20% additive oil ran 111% longer than the oil with no additive. This is still approximately a 70% reduction in rate of wear for each 10% addition of additive, although the second 10% addition only gave 111% instead of 140%. This may be discounted somewhat because as will be shown in Fig. 4, this oil was tested at a higher rubbing pressure.

The significance of these results is that the reduction in rate of wear is approximately proportional to the amount of additive, regardless of the amount of the cutting oil present (i.e.—the surface just wet with oil as one exreme to the work being flooded as the other extreme).

The total load on the tool-blade was so high on the lube-and-cool test for cutting oil C, that on the upper portion of the curve, the blade was brinelling a groove into the work piece. About 50% of the tool-blades snapped off before the test runs were completed. For that reason, a comparison of the cooling effect of the cutting oils was made at just 10 milligrams of wear; it is felt that this is a more equitable comparison. As was cited above, for 10 milligrams of wear on the tool-blade, the effect of cooling by twin oil jets was to decrease the rate of tool wear from 21% to 33%.

The areas between the two curves for each oil could no doubt be increased by a still more effective means of cooling the tool. Using higher oil jet pressures, greater volume, refrigerating the coolant, etc., would probably extend the lube-and-cool curves to the right.

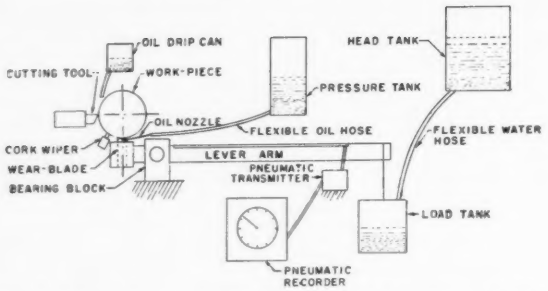


Fig. 1 Schematic of tool-blade wear test for cutting fluids.

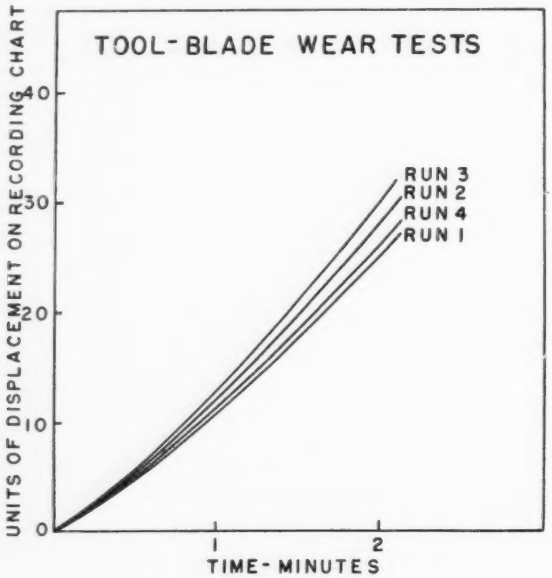


Fig. 2 A typical set of repeat runs.

This test gives an objective measure of the lubricating and cooling ability of a cutting fluid, under conditions that can be made very similar to any actual metal cutting operation. By this method, the best of several different cutting fluids can be selected for any given application.

This test does not concern itself with the practical difficulty of how the cutting fluid will get in between the chip and tool bit, but it does indicate the oil which will give you the best results if and when the oil does get in between the chip and tool bit. With a comparison of results between actual percentage improvements in tool wear and percentages as herein determined, a cutting fluid efficiency can be determined to indicate how much of the potential improvement due to a cutting oil is being obtained.

Fig. 4 shows the normal pressure between the work and tool blade for these tests. At the beginning of the test, there is theoretically only a line contact between the work and blade, so the pressure approaches infinity, but as the blade wears in a small amount, there is some area of contact so the pressure can be computed. The pressure in this figure was computed by dividing the area of the chord of the wear arc on the blade, by the total force pushing the blade against the work. Since there is some vibration between the work and blade, instantaneous pressures probably go much higher

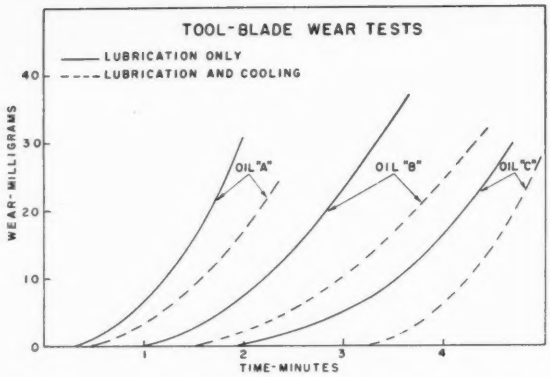


Fig. 3 Comparison of tool-blade wear for three cutting oils.

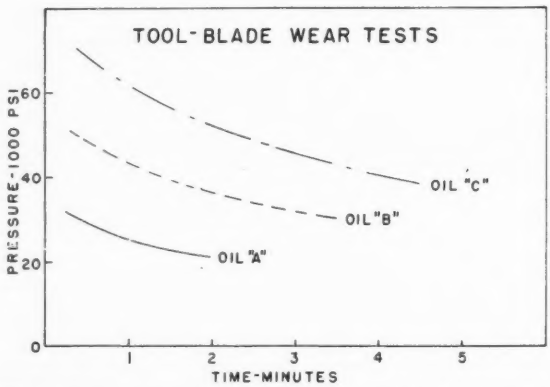


Fig. 4 Comparison of tool-blade pressure for three cutting oils.

than shown by the curves.

This figure also points out that the better the cutting oil, the more severely it is being tested, i.e. working under higher contact pressure. For example, after one minute of test, cutting oil A is under 25,000 psi. pressure, cutting oil B is under 43,000 psi. pressure and cutting oil C is under 59,000 psi. pressure. If the test procedure were changed to compare the oils at a constant rubbing load, the percentage improvements due to lubricating effect and cooling effect cited above would be higher. If a metal cutting operation were being designed for a certain tool load, this method might be more satisfactory.

The method used in this experiment is believed useful because it tends to approach what actually happens in a metal cutting operation—the poor oil will allow the tool to wear in more rapidly giving a lower chip rubbing pressure, whereas the good cutting oil will keep the surface lubricated, so that the tool does not wear in as rapidly, with the result that a higher chip rubbing pressure is encountered.

Conclusions. The results of these tests show that this tool-blade wear test for cutting fluids is a good measure of a cutting fluids lubricating and cooling ability. Nascent or freshly cut metal, heat, and pressure are all present during the test, similar to those present in most metal cutting operations. By this test, variations in the rubbing speed and tool-blade pressure can be proportioned to simulate an actual metal cutting operation. By using low speeds and high pressures, an operation such as

broaching can be simulated, and by using high speeds and light pressures, an operation such as light metal turning can be simulated. Various combinations of tool bit material and work material can be used in these tests to duplicate that being used on any actual metal cutting operation.

Modifications of this test can be made wherein the length of time may be measured to obtain a given amount of wear on the tool-blade, or the test may be run for a fixed length of time and the amount of wear determined. This can be done using a variable loading rate or a constant load on the tool-blade arm.

These tests also show that each of two 10% additions of a sulpho-chlorinated additive to a base oil could reduce the rate of wear between the chip by about 70% and that the effect of cooling by excessive cutting oil flowing over the tool is to reduce the rate of wear on the tool bit from 21% to 33%.

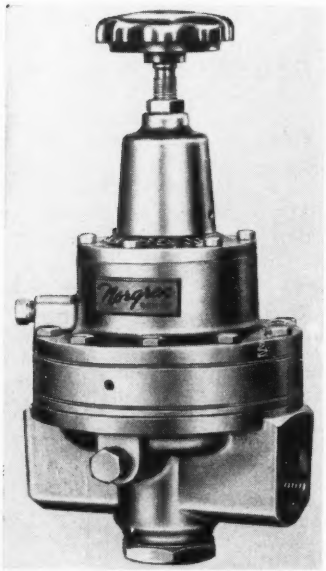
REFERENCES

[1]The results presented were obtained from tests conducted in the Mechanical Engineering Dept. of Case Institute of Technology.

[2]E. D. Tingle: Influence of Water on the Lubrication of Metals. Nature, Vol. 160, No. 4053, 1947, p. 710.

[3]The Vanton Equipment Corp., N. Y., N. Y.

Lubrication in the News



Pilot-Controlled Regulator. A new pilot-controlled air pressure regulator designed for extreme percision in air control over a wide operating range, reduces line pressures up to 400 psi to working pressures from 2 to 120 psi. Utilizing regulated air pressure to control delivered air pressure, a light adjusting spring makes regulation easy and precise and eliminates the heavy adjusting springs of other type regulators; balanced valve construction provides more accurate pressure control and greater air capacity (cfm). Available in ½," ¾", and 1" sizes, a companion regulator with the same operating features is available with a remote control unit. The regulator can be located at any point in air system regardless of accessibility, and the Pilot Regulator installed at the most convenient location. For further information, write: C. A. Norgren Co. (LE10/1), 3434 S. Elati, Englewood, Colo.

Sperm Oil & Sperm Oil Products Bulletin. A new technical bulletin on Sperm Oil and Sperm Oil Products has been issued in handy folder form, covering characteristics, chemical and physical properties, and composition of twelve oils and waxes as well as solubility data. Thirty-five of the most common uses in many industries from metalworking to textiles are included. Write for Bulletin No. 904-A, Archer-Daniels-Midland Co. (LE10/1), 2191 W. 110th St., Cleveland, Ohio.

The Coloroute Plan. A new method of completely controlling every phase of industrial lubrication makes easy the identification and segregation of lubricants in the oil room, prevents mis-application of lubricants in the plant; definitely assigns lubrication responsibilities, reduces the number of lubricants, provides "barrel to bearing" application to deter contamination, highlights hidden lubrication points, and greatly simplifies the oiler's job. The Plan originates in the oil room where each lubricant drum is color coded, together with application units and each machine bearing in the plant similarly color coded. To do the job, a Coloroute Kit containing quick-drying lacquer in six colors in as many spray containers, a set of stencils, each of different design to represent the varying frequencies of applications of different oils or greases, plus lubrication analysis sheets, master record cards, and an oil room identification chart, are included. In the oil room the oiler transfers lubricant from the color marked sealed drum to the same color marked lubrication gun; in the plant, the gun is used only to lubricate those bearings marked with the same identifying color; frequency is determined by the shape of the symbol on the bearing, insuring the right lubricant for the right bearing at the proper time. For further details, write: Alemite Div., Stewart Warner Corp. (LE10/1), 1828 W. Diversey Blvd., Chicago 14, Ill.

Lubriflush Bulletin. Comparisons of methods for lubricating electric motors are depicted in a new bulletin illustrating the three principal types of bearings used in motors: cartridge, standard open, and Lubriflush types; with explanations of the function of each. Proof that any motor bearing breathes air and therefore leads to contamination of the bearing is demonstrated photographically; for demonstrating the action of re-lubrication in the U. S. Lubriflush bearings, a series of natural color photos of a transparent bearing housing is reproduced. Write for Bulletin No. 1848, U. S. Electrical Motors, Inc. (LE10/1), Box 2058, Los Angeles 54, Calif.

Organic Chemicals Bulletin. The '54 edition of Physical Properties of Synthetic Organic Chemicals has just been released, presenting data on more than 330 products and featuring 36 new chemicals. For easy reference, the chemicals are arranged by family groups; condensed data on applications are presented and physical properties are given in tabular form. Write for Bulletin No. F-6136, Carbide & Carbon Chemicals Co. (LE10/1), 30 E. 42nd St., New York 17, N. Y.

Beta Ray H/C Meter. Invented by R. Jacobs & L. G. Lewis, Standard Oil Co. (Ind.), the H/C meter analyzes petroleum prod-

(Continued on page 41)

THE TESTING OF USED OIL*

by C. L. Pope & D. A. Hall
Eastman Kodak Co.
Kodak Pk.
Rochester 4, N. Y.

ABSTRACT: Tests which have been found useful for evaluating used lubricants are discussed and examples given to illustrate their advantages and limitations. Simple physical and chemical tests as a class have the advantages of small sample size, speed and low cost and should be employed wherever they give adequate information. The extent of change that can be tolerated cannot be arbitrarily fixed since it varies with the oil and the application. Accelerated service tests, properly devised and employed, are very informative. They can be used to measure reserve oxidation stability and estimate unexpended service life.

The testing of used oil is an interesting and important phase of lubrication engineering. Interesting because of the great variety of things that can be found in used oil; water, rust, metal chips, soaps, algae, various fibers, solvents, thickeners, varnish, paint, oxidation inhibitors, rust inhibitors, foam inhibitors, film strength additives, pour point depressants, detergents, acids, bases, salts, and oil. Interesting too because, in the application of the theory of lubrication, it is necessary to know the limits of deviation that can be tolerated. Important because the operation of plant machinery depends upon the proper functioning of its lubricants. It is the efficient and continuous operation of machinery and not the cost of the lubricant that is so important in lubrication engineering. The cost of a few gallons of oil is insignificant when compared to the cost of down time and maintenance

The person who is going to do a good job of testing used oil must first be thoroughly familiar with the properties of the new oil. Then he must carefully note and evaluate differences between the oil when new and after a certain period of use. These differences can be caused by changes in the oil itself or by contamination. Many times used oil must be discarded because of the accumulation of some contaminant even though the oil itself is in good condition. Sometimes the presence of a contaminant in the oil indicates the failure or incipient failure of a machine part. In one instance, bronze dust in the oil from a hydraulic extrusion press, after being in service only one month, indicated the difficulty to be due to a large roller thrust bearing, because the retainer of this bearing was the only bronze part in the machine. Provision was made for an extended shutdown and a spare bearing was provided. When the press was disassembled, the bearing seat was found to be badly misaligned.

Another common contaminant is excessive water. This usually indicates a leaky oil cooler or steam gland. Freon is commonly encountered in refrigeration compressors. It causes the oil viscosity to drop markedly. While a little Freon in this type of oil is normal and acceptable, larger amounts call for some attention to the seals. There are many contaminants and many ways of detecting them. They cannot all be enumerated here. In general, we learn by experience how much of a given contaminant can be tolerated in an oil in a given machine.

When we come to study the changes which occur in the oil itself as it is being used, we enter a field that is vastly more complex. Most oils today that are tested as used oils contain inhibitors of one sort or another. As long as the inhibitors are in good working order, the oil is in good working order. When the inhibitors are depleted, the oil deteriorates rapidly. Unfortunately, we do not have simple chemical tests available to tell us how much of each inhibitor is left in a used-oil sample. We don't want to wait until the oil breaks down in our plant machinery to find out when the inhibitor is depleted because then we would have to shut down the machinery and clean up the mess. But if we could know ahead of time just when the oil was going to break down, we could arrange a convenient time to drain the old oil and refill with new.

Accelerated service tests have been devised that will break down an oil 10 to 100 times as fast as a given piece of plant machinery. An acceptable accelerated service test must only intensify the factors causing breakdown of the oil without changing the essential pattern of the breakdown, or the products that are formed, from that which occurs in the plant machinery. Heat, load, speed, oxygen, and metal catalysts are used in accelerated service tests. When a good test has been found, the deterioration pattern of an oil in it is recorded. Periodically, a sample of the used oil is withdrawn from the machine and tested in the same manner. When this pattern is compared to that of the original sample, an estimate can be made of the remaining useful life of that oil in that machine. It is very important to compare the properties of the used oil with those of the original sample and this point should be kept constantly in mind. However, it is not always necessary to run the complete accelerated service test on used oils. Once the correlations between test hours and service hours has been established, it is only necessary to run as much of the accelerated service test as will assure satisfactory performance for an additional reasonable period, such as six months or a year, in the plant machinery. Testing in this manner every six months will give adequate protection against oil breakdown.

*Sponsored by the ASLE Technical Committee on Physical Properties of Lubricants, and persented at the ASLE 8th Annual Meeting, Boston, April 15, 1953.

The accelerated service test should be considered the main source of information on the condition of a used oil. However, since it is time consuming, one is often forced to glean as much information as possible from the standard physical and chemical tests for lubricants and these are discussed below.

Viscosity. The viscosity of a well refined mineral oil changes very little during its useful life. Usually a range of plus or minus 15% of the viscosity of the new oil will cover its useful life. An increase of more than 15% indicates extensive oxidation or the presence of a vegetable or animal oil. A decrease in viscosity usually indicates dilution with a solvent.

Acid Number. It does not appear possible to give a limiting acid number that will apply to all lubricants[1]. Acid number does indicate that a change is occurring in the lubricant, and is an easy value to obtain. Some lubricants are unfit for service with an acid number as low as 0.20, and others may be serviceable as high as 2.0 or higher. Acid number may be used with reliability provided one has learned from experience with a particular lubricant or has run an accelerated test. The curve in Fig. 1 shows the widely varying acid numbers of two different turbine oils at the time they laid down deposits in the oiling system of a steam turbine. It is difficult to use the acid number of lubricants containing an oxidation inhibitor as a guide for the condition of the oil. These oils will show no significant change in the acid number as long as the oxidation inhibitor is effective. Once it is no longer effective, and the acid number rises, further useful life may be extremely short, as such lubricants have generally been drastically refined to make them more susceptible to the inhibitor.

One may be lulled into a false sense of security by following acid numbers in a system lubricated with one of these oils by neglecting also to make routine accelerated tests. One example should suffice. A system holding one hundred gallons was oiling four heated bearings. The oil leaving the bearings was 380 F., after which it was cooled and recirculated. Weekly samples were taken and no significant change in acid number was noted after four weeks. During the fifth week the oil deposited such heavy resins in the bearings that the machine "set-up" while in operation.

Interfacial Tension. Interfacial tension between an oil and water is an extremely sensitive test for hydrophilic compounds in the oil. These are usually the acidic components and polar compounds. A high interfacial tension usually indicates that the oil has little affinity for metals. Thus a white oil with an interfacial of about 55 dynes per centimeter fails to wet metals well and, therefore, is a poor lubricant for bearings. A low interfacial tension indicates good wetting properties, possibly good detergency, but poor demulsibility. A combination of a high interfacial tension in an oil when new followed by a marked decrease in use, denotes oxidation or breakdown of the oil. Cooperative testing by the utilities and the oil producers has shown that when the interfacial tension falls below 15, a transformer is definitely sludged. When it is between 20 and 15, the transformer may or may not be sludged.[2]

Some oils containing rust inhibitors have a rather low interfacial tension when new and experience

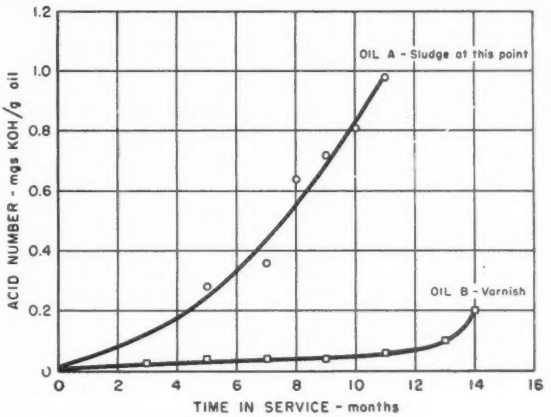


Fig. 1 Acid number vs. service life of turbine oils.

a rise in the interfacial tension as the oil is used due to the plating out of the rust inhibitor on the metal surface. Fig. 2 shows four typical types of interfacial tension curves during the useful life of lubricants. An accelerated oxidation test is useful in determining the type of curve for a given oil so that proper consideration may be given to this property of a used oil. Where applicable, the interfacial tension, by changing markedly during the life of a lubricant, seems to be easier to follow than the acid number. The latter, in some cases, hardly changes more than the expected error in the test method during the life of a lubricant. The very small sample (10 ml) required for the interfacial tension test is sometimes of value when following the life of an oil in a small system. The amount of make-up oil added to a system is very important in evaluating the performance of an oil.

Solvent Extraction. Solvent extraction is frequently used in the examination of used oils and sludges. Many of the products of oxidation are oil soluble. These are broadly classified as asphaltenes and resins. Washing and filtering the oil with naphtha or pentane will throw these out of solution and leave the asphaltenes, resins, carbon, and metallic soaps in the filter. Washing the filter cake with benzene will remove the resins and asphaltenes. This leaves carbon, metallic soaps, and inorganic matter on the filter. By weighing between washings, the percentage of the various components may be determined. The percentage thus soluble in benzene is considered a measure of the deterioration of the oil in service from oxidation. There are no hard and fast rules for the limits permissible for oils in various types of service. According to Georgi[3], about one-half of one per cent resins plus asphaltenes is all that should be allowed to accumulate in a used crankcase oil. The value of used oil analysis for engines may be illustrated by the example in Table I. The low interfacial tension, as well as the sulphated residue, indicates that this is a detergent or so-called heavy duty oil. The addition of make up oil and the continued presence of the detergent does not make the interfacial tension of significance in this case. However, the high resin content in the used oil of engine number 2 indicates difficulties in the engine. Engine number 2 was supercharged and had inadequate oil cooling to the pistons and insufficient volume of scavenging air. The remedy

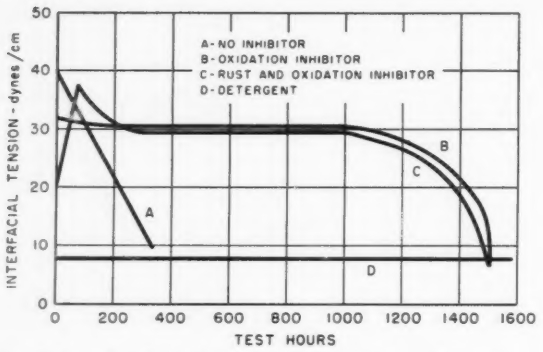


Fig. 2 Interfacial tension vs. service life of turbine oils.

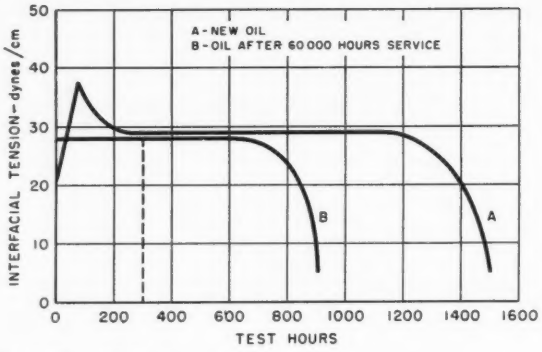


Fig. 3 The 300 hour test for used oils.

was to lower the load on the engine or increase the cooling oil and air. An early determination of the engine condition from the oil analysis prevented a major fouling up of the engine and extensive damage. As in so many cases, when already using the proper type and quality of oil, there is no use trying different oils when there is serious mechanical or design trouble.

There are some data[4] to indicate that, in oil systems not subject to fuel and other contaminants such as turbines, hydraulic equipment, etc., the oil should be discarded before the resin content of the used oil exceeds 0.05% to 0.10%.

Ash Content. The ash content of a new oil is usually a measure of some of the additives. The increase in ash content of a used oil over the new oil measures the inorganic matter that has accumulated in the oil system. The nature of the ash has a great deal to do with the amount that may be tolerated before changing the oil charge. An analysis of the ash will indicate whether or not it is mostly iron, copper, lead, tin, silica, etc. If of the softer metals such as lead, considerably more can be tolerated than if it is iron. Normally for the average gear case or circulating system, 0.10% ash is considered a high figure. However, in internal combustion engines, one to two per cent is not unusual. The reason that this high figure can be tolerated is that the ash will usually be composed of about four-fifths lead and the balance metals, from wear and silica. If non-leaded fuel is used, of course, one would consider about 0.20% as the end point of the oils service unless the contaminants are removed.

Precipitation Test. The precipitation number is interesting as it is a measure of all of the oil insoluble materials in a lubricant. If one has a precipitation number higher than 0.10%, the precipitate should be analyzed to determine its nature. If organic, it is probably lint, paper, fibers, etc. In a splash lubricated gear case, one could permit a fairly high concentration. However, in a circulating oil system, the concentration would have to be low to prevent damage due to plugged oil lines, strainers, restricted orifices, etc. In oil systems consistently showing a high precipitation number, a more effective means of keeping the contaminant out should be developed or filters should be used if contamination is unavoidable.

Flash Point. The flash point of used oils is of some interest in certain types of equipment. Very low flash in used internal-combustion engine-oils indicates fuel dilution, and definite increase in the hazard of crankcase explosions. Unless it is suspected that an oil in service has been contaminated with a solvent (in which case the viscosity would be affected) it is not believed that the flash point is of any interest.

Pour Point. The pour point of used oils invariably rises. If the lubricants are used in locations where the pour point is of interest the oil will have to be discarded when the oil reaches a maximum for the service. Ordinary filtering of a used oil does not usually restore its original pour point. This characteristic of used oils has caused difficulty in gear cases exposed to winter conditions. No general rule can be laid down as to how high the pour point can be allowed to rise in a used oil. Usually a rise in the pour point of as much as fifteen degrees F will be accompanied by other undesirable changes in the characteristics of the oil.

Rusting. Oils containing rust inhibitors tend to lose their ability to protect metal from rusting either in service or long time storage. This is believed to be due to the plating out of the rust inhibitor on the metal surfaces. This is of interest if the oil is removed from a system for drastic cleaning of the system or if new parts are put in the system. All enclosed oil systems subjected to rather quick changes in temperature are likely to condense moisture. It is not believed that it is altogether necessary that used oils pass the ASTM test D 665-46T for rusting. Certainly, the make-up oil being added should pass this test and the oil should be changed if the system is cleaned.

Oxidation Test. There is no standard, published, test-method for determining the remaining effectiveness of an oxidation inhibitor in a used oil. This is unfortunate as it is of vital importance in arriving at the remaining usefulness of the oil. Referring to Fig. 2, it will be noted that oils containing an oxidation inhibitor have a long flat curve for interfacial tension before it drops rapidly to failue. The problem with a used oil is to determine where it is on this flat portion of the curve. Sometimes, at rather long intervals, say yearly for a high temperature steam turbine, the used oil is put into an accelerated oxidation test and the life of the used oil is compared with the life of the new oil. It is not necessary to run the oil for the total accelerated oxidation test as, after a number of hours, one knows that the oil is satisfactory until the next sample will be taken. Fig. 3 shows the oxidation curve for a new steam turbine oil, and a used oil for three hundred hours, In our test[5], three hundred hours of test represents

over 11,000 hours of operation of a high temperature turbine.

Mild Extreme Pressure and Extreme Pressure Lubricants. It is neccessary to know whether or not an extreme pressure lubricant retains its so-called load carrying capacity in service. One may always make a comparison of the used oil on the test equipment used to determine its e.p. properties, i.e., Alman, Timken, SAE, Falex, Shell Four-Ball, etc. If the additive is known, such as a lead naphthenate or oleate, correlation can be obtained between the percentage of lead in the used oil and the test machine from which a suitable figure can be derived for discarding the oil. No standard data seem to be available as to the true significance of these tests. The lack of reproducibility when testing the same lubricants in different laboratories, or different operators or machines in the same laboratory, prevents standardizing limits for the various types of oils.

Foam, Emulsion and Demulsibility Tests. There is a published ASTM test for the foaming characteristics of crankcase oils, but none for industrial oils. The authors have only had one experience with a used oil foaming that was not due to mechanical conditions of the equipment. In this case, a turbine oil began to foam badly. The foam built up to several feet deep and about 29 gallons was lost out the vents in 24 hours. The foaming was easily stopped in a matter of minutes by adding .003% of a silicone to the system. For some unknown reason, the oil that was foaming in the turbine passed the foaming test for crankcase oils and the oil that did not foam failed to pass. It is not believed that this test should be applied to used industrial oils. Samples of foaming turbine grade oils are wanted[6], so that a suitable test may be developed.

Demulsibility or rate of separation of water and oil has only the steam emulsion test D 157-36 of the ASTM as a method. This method has shown lack of correlation and has found some disfavor. A reliable method is needed so that used oils can be tested for this property to avoid emulsion difficulties. The following is an example of the lack of correlation.

	Oil A		Oil B	
	New	Used	New	Used
Steam Emulsion No.	92	486	415	434
Service History	Complete separation		Permanent emulsion	

These were turbine grades of oil used in a 600 HP reduction gear, with about one quart of water condensing in the gear case each 24 hours. With oil A the water could be drawn off daily. With oil B the oil level built up until the system had to drained. It is unfortunate that we cannot anticipate these conditions. It must be remembered that, in affording us lubricants to protect against rusting (a far worse condition than emulsions), we are increasing the liklihood of emulsions. Some industries, notably the steel industry, have very difficult emulsion problems which are generally due to contamination by palm oil or other compounds used in the processing of steel.

Grease Tests. We are not aware of any significant tests for used greases. In most applications, a visual inspection will determine the condition of the grease. However, when greases do not perform as expected, one should determine whether contaminants or operating conditions are responsible.

	New Oil	Sample engine 1 3200 hrs service	Sample engine 2 1340 hrs service
Vis. 100 F.	987	1016	1030
Vis. 210 F.	77	78	78
I.F.T.	3.5	4.1	5.1
Resins & Asphaltenes	None	Trace	0.78
Dilution	None	Engines running on propane	None
Sulphated residue	0.35	0.30	0.29

Table I Analysis of internal combustion engine lubricant.

Conclusions. The authors have attempted in this brief paper to cover in the space allocated those tests which have proved of value in their experience for the evaluation of used lubricants. Undoubtedly many tests of interest to various industries have been omitted. The ramifications involved in the testing of used lubricants together with the interpretation of the data cannot be adequately covered in a single paper. If we can give a sense of appreciation to lubrication engineers that we can find out a great deal about the operating condition of our equipment from a used oil, which substitutes facts for opinions, we have accomplished our purpose. We appreciate the opportunity of presenting this paper, and hope that it stimulates many more on a controversial subject.

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[3]C. Georgi: Motor Oils and Engine Lubricants. Reinhold, New York, 1950.

[4]Sinclair Refining Co. Laboratory Report to Committee C-III of ASTM Committee D-2 of 5/18/51.

[5]C. L. Pope & D. A. Hall: E. K. Oxidation Corrosion Test. ASTM Bulletin No. 121, March, 1943.

[6]E. A. Rheel, General Electric Co., Schenectady, N. Y., personal communication.

COMMENTARY by H. Muller, Shell Oil Co. As laboratory testing in relation to service performance is a controversial subject, even on new oils, divergence of opinion on used oils has been, and will continue to be, wide-spread. From a practical viewpoint, therefore, it appears as though only in cases of large circulating systems and, as a result large volumes of oils at stake, the periodic analysis of used oils is warranted. In smaller systems oil changes at reasonably short intervals, to be established jointly between the machine operator, machine builder and lubricants suppliers, are preferable to the use of questionable laboratory test methods and the still more questionable interpretation of test results to determine oil drain intervals.

If used oil tests are to furnish the basis for determination of oil drain intervals, it is essential that these tests always be correlated with the tests of the new oil as, otherwise, misleading conclusions may be drawn. For example, there are today oils in use having an ash content when new of over 1% due to the very high detergent level to which they are manufactured. While used oils with such a high ash content have generally far exceeded the condemnation limit, it is obvious that this is not the case if the oil had a high initial ash content.

Regarding the authors' paper, I would like to make only very few specific comments. While it is premature to make general statements as to the value of certain used oil analyses in internal combustion engine service, latest work conducted mainly in the railroad diesel locomotive field indicates that the emission spectograph gives valuable information on the composition of inorganic matter in used diesel engine oils such as excessive silica due to inefficient air cleaner operation or the presence of high amounts of lead due to incipient bearing failure. The blotter test and inspection of the oil spot on filter paper gives useful information on the residual detergency of locomotive diesel engine oils. It seems as though these findings in the field of railroad diesel engines

may be applied to certain other fields of heavy duty engines.

The "things" to which the authors refer as being found in used oil, I believe fall into three classes. Some of the things found in used oil, of course, are the additives present in the new oil. These are very desirable things. On the other hand, one finds contaminants which entered the oil during service which are undesirable, and finally, one finds chemical compounds of deterioration in the oil which, if excessive, are also undesirable. The load on the lubricant could be greatly reduced if percautions were taken to cut down contamination. This is particularly true in cases of grease in which a recent survey indicated that the most important single cause of grease failures was contamination with such matters as water, mud, road dust, coal dust, sand, various chemicals including perioxides, nitric, sulphuric, and acetic acids, certain gases, sewage contamination, fruit juices, etc.

AUTHORS' CLOSURE: Mr. Muller has called attention to two important considerations. The first involves the economics of trying to evaluate the used oil in a small system. The authors would certainly agree that in one sense it is uneconomical to spend fifty dollars for an analysis of a few dollars worth of oil. It might be well to point out that the testing of used oil often produces information about wear or contamination that indicates the need for attention to the equipment. There is, of course, no substitute for judgment. A recent typical example is as follows: Three two hundred horsepower ring-oiled bearing motors for driving fans were experiencing bearing failures. Oil changes had been reduced to a weekly basis, at an annual labor cost of $780, yet the oil only amounts to $19.50 a year. An examination of the oil indicated that there was nothing wrong with the lubricant. Poor bearing grooving was responsible for the failures. This may seem an extreme case, but from many years of experience it is not. It is so easy for someone to say, "Change the lubricant more frequently." The total cost of such a philosophy runs very high in all our industrial plants. A few thousand dollars spent in analytical work, to give facts and confidence to production departments, pays dividends.

The second point advanced by Mr. Muller concerns some very interesting work being done with the emission spectroghaph for diesel engine oils. This is a good example of constructive thinking, and is very much in accord with our own ideas on how to use analytical tools. We did not include a discussion of spectrometric methods because it is a relatively new field; the apparatus and skilled technicians are not generally available. It is anticipated that very interesting papers will be forthcoming on this subject in the next few years. One example that we think is of interest was the investigation of the complaint that our grease which was added to the grease put in the traction motors of a diesel electric locomotive by the manufacturer was leaking out of the bearings and causing commutator trouble. An infra-red curve showed it to be the manufacturer's grease that was migrating from the bearing housing. Had we cleaned out the bearings, put in the manufacturer's lubricant, we would only repeat the trouble and waste about a thousand dollars. It is a strange thing that whenever a lubricant is changed and difficulty is encountered, the last lubricant in is always responsible in the eyes of operating personnel.

Patent Abstracts

Prepared by **Ann Burchick** from **OFFICIAL GAZETTE,** Vol. 675, Nos. 3, 4; Vol. 676, Nos. 1, 2, 3, 4; Vol. 677, Nos. 1, 2, 3, 4, 5. Printed copies of patents are available from the Patent Office at twenty-five cents each. Address the Commissioner of Patents, Washington, D. C. for copies and for general information concerning patents.

Lubricating Greases Thickened With Calcium Silicate, Patent #2,656,315
by G. W. Eckert, assignor to The Texas Co.

A lubricating grease comprising an oleaginous liquid as the predominating constituent, and about 5 to 45 per cent by weight based on the weight of the grease of a precipitated hydrated calcium silicate having a particle size of not more than 100 millimicrons in diameter.

Lubricating Oils Containing Metal Salts of a Condensation Product of a Substituted Phenol and a Vinyl Compound, Patent #2,656,316
by L. L. Neff, assignor to Union Oil Co. of California

A mineral lubricating oil containing between about 0.05 and about 20% by weight of an oil-soluble metal salt of the acidic condensation product of a hydrocarbon substituted phenol, in which the hydrocarbon substituent contains between about 4 and about 30 carbon atoms, and a vinyl compound selected from the class consisting of divinyl ether, vinyl ethyl ether, vinyl propyl ether, vinyl esters of hydrohalogen acids and vinyl esters of fatty acids containing between 1 and 6 carbon atoms in the fatty acid radical, said condensation product being obtained by condensing said phenol with said vinyl compound in the molar ratio of between 0.5 and 3 to 1 at temperatures between about —20 C. and about 150 C.

Lubricating Oils Containing Metal Salts of a Condensation Product of a Substituted Phenol and an Allyl Compound, Patent #2,656,317
by L. L. Neff, assignor to Union Oil Co. of California

A mineral lubricating oil containing between about 0.05 and about 20% by weight of an oil-soluble metal salt of the acidic condensation product of a hydrocarbon substituted phenol, in which the hydrocarbon substituent contains between about 4 and about 30 carbon atoms, and an allyl compound selected from the class consisting of allyl alcohol, diallyl ether, allyl ethyl ether, allyl propyl ether, allyl esters of hydrohalogen acids and allyl esters of fatty acids containing between 1 and 6 carbon atoms in the fatty acid radical, said condensation product being obtained by condensing said phenol with said allyl compound in the molar ratio of between 0.5 and 3 to 1 at temperatures between about —20 C. and about 150 C.

Mineral Oil Additive, Patent #2,656,-318
by J. H. Bartlett, assignor to Standard Oil Development Co.

A mineral lubricating oil containing dissolved therein 0.01 to 15% by weight of a product obtained by reacting a combination of the elements phosphorus and sulfur selected from the group consisting of sulfides of phosphorus and mixtures of elemental sulfur and phosphorus with a polymeric material of molecular weight from about 1000 to about 50,000, selected from the group consisting of (1) polymers of an ester of a dicarboxylic acid which contains a double bond in an a,B position and which contains a total of two to four carbon atoms exclusive of the carboxyl groups, such ester having as esterifying groups aliphatic hydrocarbon radicals each containing 6 to 22 carbon atoms, the average number of carbon atoms in the radicals being 8 to 18, and (2) copolymers of an ester so defined and a compound.

Manufacture of Antifriction Bearing Greases, Patent #2,657,180
by L. W. Sproule and J. S. Gray, assignors to Standard Oil Development Co.

A lubricating grease composition comprising lubricating oil having a viscosity between about 35 and 500 SSU at 210 F., thickened to a grease consistency with a soda soap of a mixture of commercial fats and substantially saturated fatty acids, said mixture having an iodine number not greater than about 25 and being substantially free of uncombined glycerin prior to saponification, said grease being characterized by a smooth structure and short fibre imparted thereto by cooling with mechanical working to a temperature not greater than 130 F. and by introducing extraneous glycerin during such cooling and working sufficient in amount to make the total glycerin content equal to about 0.8 to about 1.5 times that of said fats and the corresponding triglycerides of said fatty acids.

Hydrocarbon Oil Containing S-(Sulfurized Terpene Hydrocarbon) Dithiophosphoric Acid Triesters, Patent #2,657,900
by E. O. Hook and L. C. Beegle, assignors to American Cyanamid Co.

A hydrocarbon oil composition comprising a major amount of a hydrocarbon oil having dissolved therein a minor quantity, sufficient to inhibit oxidation of the oil, of an S-(sulfurized terpene hydrocarbon) 0,0-dihydrocarbon dithiophosphoric acid triester.

Manufacture of Lubricating Greases, Patent #2,658,037
by E. Orno-Ornfeldt, assignor to

(Continued on page 45)

AIR BEARING STUDIES AT NORMAL & ELEVATED TEMPERATURES*

by J. D. Pigott, Osborn Mfg. Co.
Cleveland, Ohio
E. F. Macks, Engrg. Consult.
22758 Maple Dr., Cleve., O.

ABSTRACT: Experimental studies were made with a six-inch outside diameter, externally-pressurized, parallel surface, nonrotating, air thrust-bearing at temperatures to 1000 F. Theoretical expressions describing the air flow through the bearing and the air flow through capillary tube resistances are presented in the form of design equations and curves and are compared with experimental results over a wide range of temperatures and loads. Results show that the load capacity is increased as the operating temperature is increased.

Current design trends in many fields of bearing application demand that bearings operate effectively at higher and higer temperatures. The temperature limitations of existing petroleum and synthetic lubricants and the lack of adequate high temperature materials place a limit on maximum bearing operating temperature. In certain other applications at normal temperatures, the demand for extremely low coefficients of friction presents still another problem for which conventional rolling contact and oil-lubricated hydrodynamic and hydrostatic bearings are inherently unsuitted. The use of air as a lubricant presents one possible approach to these problems.

Considerable work has been done in the past on air bearings, with Hirn[1] first postulating in 1854 that air might be a very desirable lubricant. In 1897, Kingsbury[2] conducted experiments in which he successfully operated an air-lubricated journal under hydrodynamic conditions. Harrison[3] developed theoretical expressions in 1913 which described the hydrodynamic characteristics of a compressible fluid, but it has since been realized[4] that the viscosity and compressibility of air limit its use under hydrodynamic conditions except under extremely light loads. The trend in air bearing design has thus been to employ the hydrostatic principle of lubrication[5,6] wherein an externally pressurized source of air is used to supply the pressure necessary to support the applied load. Several ingeneous methods have been used [7,8] in bearing designs utilizing the hydrostatic principle in specialized applications at normal temperatures. Theoretical information in the literature is scanty and is devoted mainly to these normal temperature, specialized designs. The work of Midwood and Duncan[9] and Shires [10,11] provides a comprehensive background for air bearing theory and has been supplemented by their experimental work at normal temperature. It is the purpose of this paper to extend this basic knowledge in the field to elevated temperatures, where little information is available.

The parallel-surface, nonrotating thrust bearing, which was analyzed in this instance, presents the simplest possible geometry for obtaining basic flow equations as functions of temperature. This analysis is intended to provide a temperature-dependent theoretical foundation which may be refined eventually to include more complicated geometrical and rotational effects. The effect of rotation was not studied in this work, but preliminary tests at normal temperatures by other investigators[11] indicate that relative motion of the surfaces has little effect upon the statiic characteristics of an externally pressurized air bearing over a large range of speeds.

ANALYSIS. The bearing analyzed was of the type shown in Fig. 1. The load W, applied to the top plate through the bearing at E, is supported by the externally pressurized air film between plates. The four capillary tubes through which the inlet air enters the four recessed pads in the bearing surface are shown at A. The air flows from a common manifold at pressure p_m, through the four external resistance tubes to the pads in the bearing surface in which the pressure is p_s, related to p_m by the pressure drop in the tubes for any given mass flow. The air then escapes rapidly to atmosphere p_a through the annular clearance surrounding the pads, providing the load-carrying air film as it does.

One of the greatest difficulties with air bearings has been their tendency toward instability in general due to such specific factors as misalinements or variations in applied load. The use of external resistance tubes in this design is one means of minimizing these instability effects. When manifold pressure p_m, is maintained constant, pad pressures p_s, will vary, collectively or individually, as the flow through the external resistance varies. With references to Fig. 1, if the upper and lower bearing plates become misalined so that the film thickness between plates in the vicinity of B is greater than the film thickness diametrically opposite at C, the flow through resistances 1 and 3 will be greater than that through 2 and 4. Thus, since the pressure drop in each resistance varies directly with flow, the pressure p_s in pads 1 and 3 will fall below that in 2 and 4. This readjustment in pressure causes the plates to stabilize toward their original parallel position. Secondly, if the bearing undergoes a small variation in load W a stabilizing effect is produced. If the load decreases, the plates tend to separate, but this increase in film thickness produces an increase in flow through all four resistance simultaneously, producing a drop in pressure p_s in all four pads. As a result of this over-all drop in pad pressures, the plates tend to stabilize toward their original film thickness.

*Sponsored by the ASLE Technical Committee on Bearing Lubrication, and presented at the ASLE 8th Annual Meeting, Boston, April 15, 1953.

Other successful methods have been employed for providing the manifold-to-pad pressure differential supplied in this design by the external resistance tubes. Among them are the nozzle-type inlet of Mueller[12] and the annular choked inlet of Duncan[9]. The capaillary tube external resistances used in this design allow the dissipation of energy over a long length of tube so any desired pressure drop may be accomplished at minimum flow velocities. The possibility of inlet turbulence and shock and resultant bearing instability is thereby minimized. It is realized that disadvantage are inherent in the use of capillary tubes, amoung them the possibility of clogging and added design complication. However, for this work, extreme stability was desired and a study of capillary tube flow, on which data is limited, was of interest.

The analytical summary is divided in two sections: (1) flow equations for capillary external resistances and (2) equations for parallel-plate flow between the air bearing surfaces; this flow is in a radial direction. As implied above, these analyses are related by the fact that the mass flow between the bearing surfaces (Q) equals the total mass flow through the external resistances ($4Q_t$).

Flow Equations for External Resistances. Several flow equations are available for fluid flow through tubes and pipes and a brief summary follows. More complete analytical details are available in Refs. 13, 14, 15, 16, and 19. A list of symbols is given at the end of the text.

For incompressible laminar flow through tubes of small diameter, the most commonly used expression is the Hagen-Poiselle Law:

$$Q_t = 3600\,\pi d^4(p_m - p_s)/128\,\mu L \qquad (1)$$

If the perfect gas law $p = \rho RT$ is introduced into the Hagen-Poiselle analysis before integration, the following equation for compressible laminar flow results:

$$Q_t = 3600\,\pi d^4(p_m^2 - p_s^2)/265\,\mu RT_t L \qquad (2)$$

Using an analysis upon change in momentum and a friction factor such as that defined in Ref. 13, turbulent flow is obtained.

$$p_m - p_s = \rho f' LV^2/d2g = Kf' LQ_t^2/\rho d^5 \qquad (3)$$

When the momentum-friction factor analysis leading to Eq. (3) is extended by use of the perfect gas law, a relation valid for compressible or incompressible, laminar or turbulent flow is obtained.

$$p_m^2 - p_s^2 = 20.8 \times 10^{-8} T_t f' LQ_t^2/d^5 \qquad (4)$$

The details of the derivation of Eq. (4) from basic theory have been deleted in the present paper but the equation based on this derivation is plotted for design conditions in Fig. 2.

Radial Flow Outward From a Point Source Between Parallel Plates. A complete derivation of the equations, describing the parallel-plate flow through the air bearing and involving Reynolds number considerations, leads to the following basic design equations:

$$h^3 = \frac{12\mu RTQ\,\ln(r_a/r_s)}{\pi(p_s^2 - p_a^2)} \qquad (5)$$

$$W = \pi r_s^2 p_{sg} + \pi p_{sg}(r_a^2 - r_s^2)/2 \qquad (6)$$

Eqs. (5) and (6) are valid for compressible laminar flow radially outward between flat parallel plates under the assumptions used in the derivation of these equations. A load capacity analysis has also been developed which leads to a complex integral which has been integrated by numerical means.

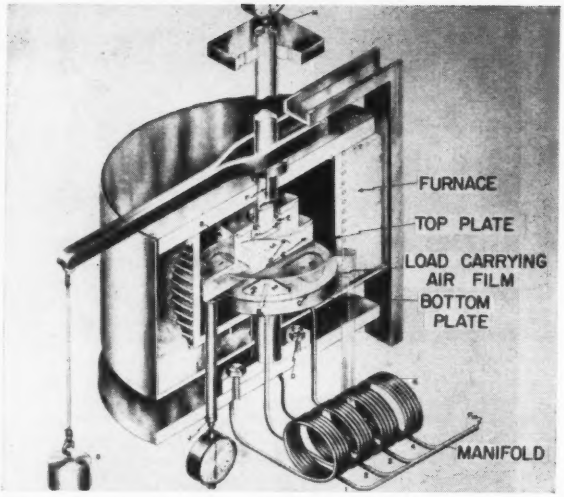

Fig. 1 Apparatus used for testing four-pad air bearing at temperatures up to 1000 F.

When this numerical integration was carried out it was found to correspond, with negligible error, to the assumption of a linear pressure drop from p_s to p_a. Eq. (6) is the simplified load capacity equation obtained by assuming the linear pressure drop from p_s to p_a.

It may be seen from Eq. (6) that load capacity W is a function of pad pressure (p_s) and geometry (r_a , r_s) only. Thus, the family of film thickness curves in Fig. 4 are obtained from Eq. (5) over a range of mass flows and temperatures at various loads for the bearing tested. The dependence of the viscosity of air on air temperature must be included in these film thickness calculations, and may be obtained from Ref. 20. The viscosity increases almost linearly from 2.7×10^{-9} reyns to 80 F. to 5.24×10^{-9} reyns at 1000 F. The effect of pressure on air viscosity has been found negligible.[3]

APPARATUS & PROCEDURE. The details of the apparatus may be observed by referring once again to Fig. 1. The bearing was made of Inconel and its surfaces lapped to 10 microinches rms. The radius of the pads (r_s) was 2.5 inches and the outside radius of the bearing (r_a) was 3 inches, providing a ½ inch length for escape of air from p_s to p_a. The ribs in the bearing surface which separated the four pads were ½ inch in width. The pads were very shallow (0.005 in. deep) in keeping with the theory of Mueller[12] for added stability against expansion shock.

Initial alinement of the plates was obtained by adjusting three screws at 120° intervals around the base of the bottom plate as shown at D until the pad pressure were equal and film thickness uniform at the bearing surface. With the test setup used, the top bearing plate was free to aline itself continuously due to clearance in the ball bearing at E.

The external resistances were 0.040 inch I. D. seamless Inconel tubing 10 feet in length. These tubes were wound compactly into a coil package 4 inches in diameter (A in Fig. 1) with negligible effect on friction factor values compared to those for a straight length over the flow range tested. Inlet air was filtered to a maximum particle size of 2 microns and dried to a maximum relative humidity of

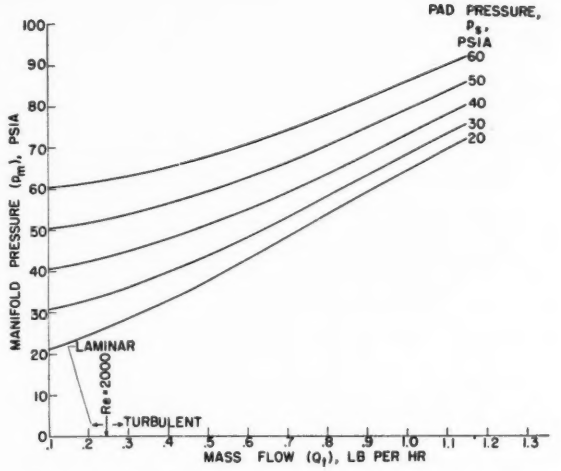

Fig. 2 Manifold pressure versus mass flow through capillary tube based upon equation (4) with T = 540° A = 80° F; L = 120 inches; d = 0.040 inches.

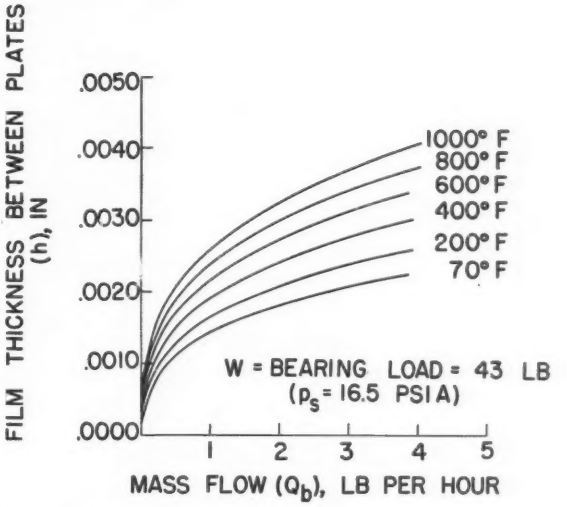

Fig. 3 Theoretical film thickness versus mass flow for the four-pad bearing tested based upon equation (5).

70 percent. Manifold and pad pressures were measured to ±0.3 psi with Bourdon type gages, and flows were measured to±5 percent with a conventional float-type rotameter.

The entire bearing was surrounded by an electrical resistance furnace. Average bearing temperature was measured with four thermo-couples (one in each pad) and controlled to within ±20 F. with an automatic on-off temperature regulator. The external resistances were located outside the furnace for all tests and their temperature, t_t, maintained at 80 F. ±5 F. except for the 6 inches of their length entering the pads through the bottom of the furnace. The capillary tubes were maintained at room temperature so that the pressure drop from p_m to p_s (see Eq. (4)) was essentially constant at any given load and mass flow and did not change with changes in bearing test temperature, t.

Loads were applied by means of dead weights at G and the system precalibrated for load to ±0.06 pound. Tests were also made under the gravity load of the top bearing plate only, using three guides at 120° circumferential locations around the radius r_a for centering the plates, and it was found that the method of loading did not influence experimental results. Air film thicknesses were measured with 0.0001 inch dial indicators which were calibrated against a standard and found to be accurate within ±0.0001 inch. Film thickness measurements were taken in two locations; a single gage mounted at the top of the load spindle at H, and three gages on the bottom faces of legs J protruding through the furnace base at 120° intervals circumferentially. Values obtained by these two methods were reproducible within measuring accuracy. Gages were zeroed with the bearing plates in contact after load application and temperature equilibrium has been accomplished, thus eliminating deflections and differential temperature expansions from the readings.

Auxiliary tests were made at room temperature on a setup having a single circular pad of radius r_s with the air inlet located at the center in which the flow was truly radially outward from a point source. Tests were made under the gravity load of the balanced upper plate only, which was 10.8 pounds, and the plates were maintained concentric by guides located at 120° circumferentially. The pad recess radius (r_s) was 0.905 inch and the plate radius (r_a) was 1.079 inches. Pad depths of 0.062 inch and 0.005 inch were tested. Film thicknesses were measured with three dial indicators spaced at 120° circumferentially. Air was introduced through a single external resistance 0.040 inch in diameter and 10 feet in length. Manifold and pad pressures were measured with mercury manometers to ±0.10 inch of mercury.

ANALYSIS. Capillary External Resistance Tubes. Reference to Eq. (4) and Fig. 2 enables one to observe the design variables influencing the manifold-to-pad pressure drop. It is desirable from a stability point of view to maintain mass flow as small as possible for a given desired drop in pressure. This may be accomplished by proper choice of capillary tube length and diameter.

Friction factor values were obtained for 0.040 inch diameter seamless Inconel tubing using a simple experimental setup to room temperature which measured p_m, p_s, and Q_t. Since the friction factor varies with both tube roughness and diameter, a similar experimental setup must be employed when using tubes of other roughness or size. The values of friction factor herein found are of the same order of magnitude as those of other investigators for somewhat larger tube and pipe [13, 14]. Choice of operating pressure ratio p_m/p_s will depend upon the film thickness or the alining and load fluctuation stability desired. From a stability point of view, it is desirable to operate at a point where the p_s curves of Fig. 2 are closest together, so that a given change in mass flow Q results in a maximum change in pad pressure p_s with manifold pressure constant. Determination of optimum operating pressure ratio for maximum stability effect may be carried out, but was not included in this investigation, since variation of film thickness over a range of pressure ratios and flows was the chief concern.

Analysis of Flow Between Bearing Plates. The increase in load capacity of the air bearing tested with increase in temperature may best be observed by noting the theoretical curves of Fig. 3. Here it may be seen that with all other conditions constant

the film thickness will increase due solely to an increase in temperature. The cause of this desirable effect may be observed in Eq. (5) where it may be noted that the cube of the film thickness varies directly with both viscosity and absolute temperature.

Experimental values are compared with theoretical values in Fig. 5 for a fixed load of 70.7 pounds. It may be seen that experimental values vary as much as 30 percent from those predicted by theory and are consistently lower. The following factors contribute in varying degree to this deviation of experimental results from theory:

(a) Although the surfaces were lapped, some roughness and waviness necessarily remained through which flow took place before a finite film thickness was established. This initial flow may be observed in the curves of Fig. 5 as an intercept of the abscissa.

(b) Distortions in the bearing plates with changes in temperature may magnify or minimize the waviness effect as the test temperature is changed. Thus a change in the magnitude of the deviation from theory may be expected with temperature change.

(c) Because the cube of the film thickness is proportional to mass flow to the first power, the absolute difference between theoretical and experimental mass flow because of initial film thickness differences will increase as mass flow increases.

(d) Simplifying theoretical assumptions made in the derivation of equations (4) and (5).

It is most important in an air bearing design to realize the large deviation of mass flow from theory to which such factors as surface irregularities and temperature distortion may contribute.

Experimental values, such as those presented in Fig. 5, were determined at loads of 43, 70.7, 113, 147, 182, and 217 pounds over a range of temperature from 80 F. to 1000 F. and all exhibited the same characteristic relationship with theory as those of Fig. 5.

During the course of the experimental work, it was felt that some of the difference between theoretical and experimental results might be due to the presence of the four pads and separating ribs in the bearing surface causing the flow to deviate from the radial flow outward from a point source upon which the theoretical expressions were based. The setup having the single circular pad which corresponded more closely to the conditions assumed theoretically, was employed to investigate this further. To establish a condition wherein pressure losses within the pad area would be a minimum, and pressure p_s constant over the pad area as assumed in the theory, a relatively deep pad (0.062 in. deep) was first employed.

Results for flows up to 1 pound per hour with this setup exhibited the same characteristic deviation from theory (consistently lower film thickness values) as those for the four-pad bearing, so the effect of the pads on the flow characteristics was considered negligible. However, at flows greater than 1 pound per hour, instability and chatter made film thickness observation impossible. Since a flow of 1 pound per hour in this single-pad bearing corresponds to a Reynolds number of less than 100, it was concluded that turbulence in the flow between the plates was not responsible for the observed chatter. To determine whether the increased pad depth was the cause, the pad depth was reduced to 0.005 inch to correspond to that used in the four-pad bearing. Little change in stability conditions was observed, with chatter again occurring at flows above 1 pound per hour. Because stability was obtained at much higher flows and Reynolds numbers with the four-pad bearing having a 0.005 inch deep recesses it would appear that the instability is caused by erratic loading due to entrance turbulence and shock which is minimized by the multiple-pad, self-stabilizing design.

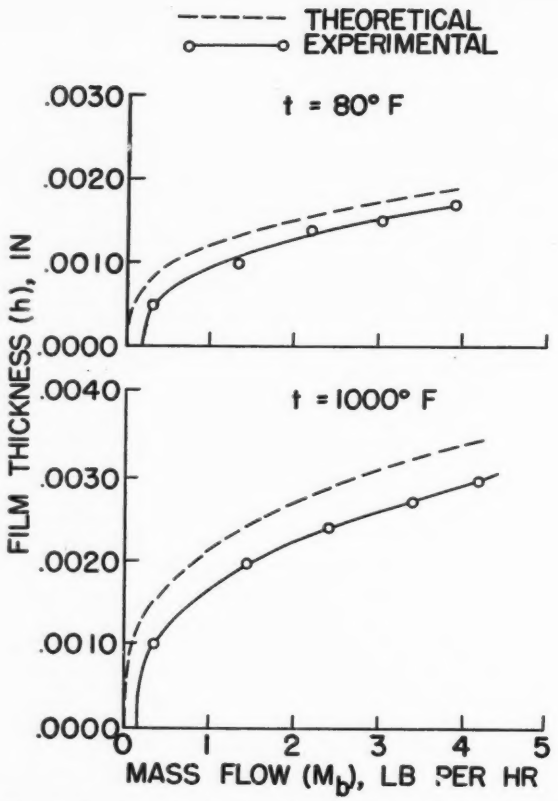

Fig. 4 Comparison of theoretical and experimental film thickness versus mass flow for the four-pad bearing tested. Applied load = 70.7 pounds. Theoretical values are based upon equation (5).

In summary, a hypothetical design problem is presented: It is desired to support a force (W) of 115 pounds on a plate whose radius (r_a) is 3 inches and pad radius (r_s) is 2.5 inches using a mass flow (Q) of 1 pound per hour. Both the capillary tube temperature, t_t, and the bearing operating temperature, t, will be 80 F. What film thickness (h) will result and what manifold pressure (p_m) is required if four capillary tubes of 0.040 inch inside diameter and 10 feet in length are used as external resistances?

Solution: The pad pressure, p_s, is obtained directly from Eq. (6) for W = 115 pounds, r_a = 3 inches, r_s = 2.5 inches and is found to be 19.5 pounds per square inch absolute. Using p_s = 19.5 pounds per square inch absolute, Q = 1 pound per hour, and T = 540° A, the film thickness (h) is found from Eq. (5) to be 0.001 inch. Substituting the previously determined values of t_t, d, L, Q_t, and p_s in Eq. (4) the required value of p_m is found to be 26.5 pounds per square inch absolute. The meas-

ured film thickness under these conditions was 0.0008 inch—20 percent lower than that predicted by theory.

For conversion of air flow from pounds to cubic feet it is noted that the density of air is 0.0765 16/ ft^3 at standard conditions (15 C., and 760 mm Hg).

CONCLUSIONS. The following conclusions may be drawn from the work herein presented:

1. Theoretical and experimental material presented herein indicate that the higher the operating temperature, the higher the load capacity of an externally-pressurized air bearing.

2. Capillary external resistances are one means of obtaining stabilizing manifold-to-pad pressure drop in an air bearing at minimum Reynolds number. Design equations for the flow through such tubes are presented.

3. Experimental results indicate that the four-pad thrust air bearing tested is capable of effective operation at temperatures up to 1000 F.

4. The degree of surface irregularities and waviness, or distortions in the bearing surface due to temperature changes, has a marked effect upon film thickness and mass flow calculations.

5. The value of the multiple-pad, self-alining design in minimizing bearing instability resulting from inlet turbulence and shock is evidenced by comparison of the four-pad design with the single-pad, pure radial flow bearing.

6. The change in stability due to a change in pad depth from 0.062 inch to 0.005 inch was negligible over the flow range tested with the single-pad, pure radial flow bearing.

NOMENCLATURE

d	diameter of tube, in.
f'	friction factor
W	bearing load, lb
g	acceleration of gravity, in/sec^2
h	film thickness between plates, in.
K	constant
L	tube length, in.
Q	mass flow through bearing, lb/sec
Q_t	mass flow per tube, lb/hr
p	absolute pressure, lb/sq in. abs
p_a	atmospheric pressure lb/sq in. abs
p_m	manifold pressure, lb/sq in. abs
p_s	pad pressure, lb/sq in. abs
p_{sg}	gage pad pressure, lb/sq in. gage
r_a	outside radius of bearing plate, in.
r_s	radius of pad, in.
R	gas constant, in lb/lb °A
T	absolute temperature of air, °A
T_t	absolute temperature of capillary tubes, °A
t	bearing test temperature, °F.
t_t	temperature of external resistance tubing, °F.
V	maximum velocity in tube, in./sec
ρ	density, lb/in.^{3}
μ	absolute viscosity, lb sec/in.^{2} or reyns

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Current Literature

Synthetic Lubricants from Dihydroxystearic Acid
by L. E. Gast, C. B. Croston, W. J. Schneider & H. M. Teeter, American Chemical Society 124th Meeting, Abstracts, Sept. 6-11, 1953, p. 91-0.

A series of derivatives of dihydroxystearic acid has been prepared and some of the viscometric properties that are of interest in evaluating these derivatives as synthetic lubricants have been determined. Pour points for these compounds were mostly in the range of —60 to —70 F. Viscosities at 100 F. varied from 15 to 30 cs.; however, at 40° viscosities were of the order of 4000 cs. or greater.

Petroleum Waxes
by S. T. Minchin, Institute of Petroleum Review, Vol. 7, No. 82, Oct. 1953, pp. 321-328.

The difference between petroleum and other waxes is that petroleum waxes are complex mixtures of paraffin hydrocarbons, whereas waxes from animal and vegetable sources nearly always contain combined oxygen in the form of acids, alcohols, esters, etc. The author discusses separation of waxes from wax distillates by pressing and sweating and by solvent dewaxing. Refined waxes categories are considered. Main uses of petroleum waxes are divided into the following categories: packaging, candle, polish, electrical, rubber, leather, and match.

Use of the Petter AVI Diesel Engine for Testing Additive Treated Oils.
by A. Towle & P. E. B. Vaile, Journal of the Institute of Petroleum, Vol. 39, No. 357, Sept. 1953, pp. 581-623.

This paper shows how the Petters AVI laboratory lubricating oil test en-

(Continued on page 42)

A NEW THEORY OF METAL TRANSFER & WEAR*

by I-Ming Feng
Massachusetts Institute of Technology
Dept. of Mechanical Engineering
Cambridge 39, Mass.

ABSTRACT: Given in this paper is a discussion of a newly developed theory of metal transfer and wear[1], which includes the mechanical interlocking effect of plastic roughening of the interface of contacting high spots as the primary cause and the adhesion or diffusion process induced by temperature flash during the breakage of high spots as the secondary cause of metal transfer and wear.

The main purpose of lubrication is to reduce friction, metal transfer and wear. When the rubbing surfaces are well separated by an uninterrupted film of lubricants under full hydrodynamic lubrication, the friction is relatively small, wear is practically nil, and metal transfer does not occur. However, in many engineering applications, perfect hydrodynamic lubrication is practically impossible. For example, under the working conditions inside the cylinders of internal combustion engines, full hydrodynamic lubrication can never be achieved. Even when it is possible to design a journal bearing to achieve full-fluid lubrication under normal operating conditions, metal transfer and wear still cannot be avoided during the starting and stopping period. This is because in rubbing two surfaces together under boundary lubrication or dry conditions, friction is generally accompanied by metal transfer and formation of loose wear particles. There is no question about the evidence of the loose wear particles, because no highly specialized technique is necessary for detecting them. The problem of detecting transferred metal has been solved by using special techniques. The observation of Sakmann, Burwell and Irvine[2] employing radioactive tracer technique is most convincing. Both metal transfer and wear cause damage to the rubbing parts and end their useful life much more often than any other form of failure. In order to reduce such damage, a good understanding of the fundamental nature of these phenomena under boundary lubrication or dry conditions is essential.

SURFACE GEOMETRY AND AREA OF ACTUAL CONTACT. To explain the new theory of metal transfer and wear, surface geometry and area of actual contact will be discussed first. If two geometrically true planes could be prepared and brought together, they would contact at all points throughout the entire surface. However, even those surfaces prepared with the best known technique are not geometrically true planes. They consist of randomly distributed peaks and valleys of varying height and depth. Actual contact between two solid surfaces can only take place at a great many scattered high spots. The summation of areas of actual contact is usually only a small fraction of the apparent area. This concept of surface geometry and area of actual contact is an important contribution made by Binder[3] and Holm and his collaborators[4].

Bowden and Tabor[5] made an extensive study of the area of actual contact. They confirmed that for flat surfaces the total area of actual contact is a small fraction of the apparent area. From the magnitudes of the normal load W and the area of actual contact A, and also from the dependence of A on W, they concluded that most of the actually contacting high spots are plastically deformed.

THE NEW THEORY OF METAL TRANSFER & WEAR[1]. Interlocking due to Plastic Roughening of Interface. There are several different mechanisms of plastic deformation of metals, all of them involving abrupt atomic movement along certain sets of crystallographic planes. When two contacting high spots deform together plastically, the abrupt movement along crystallographic planes causes roughening of the interface of contact. During any stage of deformation of a pair of high spots under normal load, their surfaces of contact must fit each other. In other words, the result of simultaneous plastic deformation yields perfectly matched roughening of the contacting surfaces. To illustrate the interlocking effect of the plastic roughening, take the slip mechanism of plastic deformation as an example. A slip plane with a slip direction lying in that plane forms a slip system. If only one system is operating, the stepwise roughening takes place only in one direction as shown in Fig. 1. It is generally the case that several systems are operating during plastic deformation, and, consequently, the roughening will take place in several different directions. The mechanical interlocking effect produced by roughening in several directions is effective in preventing separation along the original interface and thus, offers resistance to tangenttial motion.

Transferred Metal or Loose Wear Particle. When a par of firmly interlocked high spots is forced to move apart, breakage will occure at some distance away from the original interface where the section is weakest. The reason is as follows: Near the interface, the metal has been severely strain-hardened during plastic deformation. The strain-hardening effect decreases, whereas the cross-sectional area increases with increasing distance from the interface. Therefore, the weakest section is at some distance away from the interface.

Most of the energy dissipated during the process of breaking the high spots is finally transformed into heat which causes a sudden temperature flash.

*Sponsored by the ASLE Technical Committee on Lubrication Fundamentals, and presented at the ASLE 8th Annual Meeting, Boston, April 15, 1953.

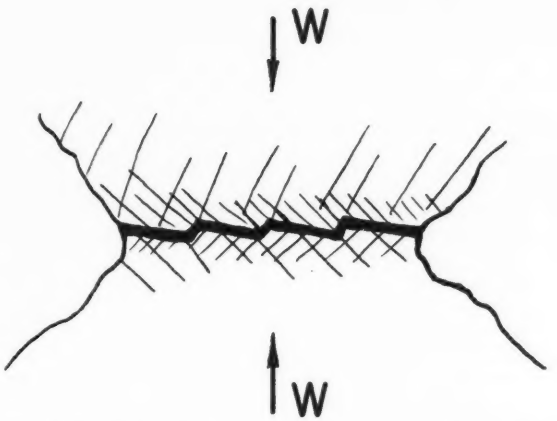

Fig. 1 Sketch showing the plastic roughening of the interface.

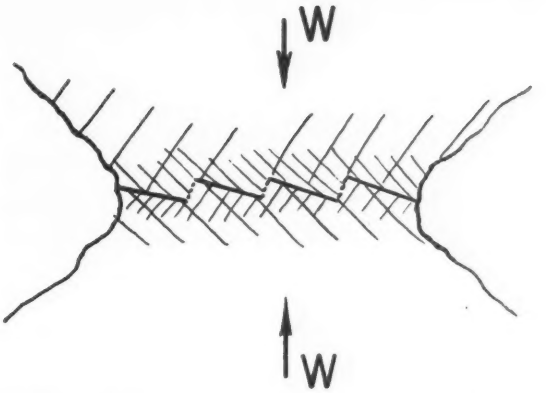

Fig. 2 Sketch showing the possible metal-to-metal contact caused by plastic roughening when the surface film is thin and brittle.

Such temperature flash has been observed by various investigators (Cf. Ref. 6). When the heat of temperature flash is conducted through the interface, momentary diffusion occurs rapidly. If the diffusion process is rapid enough to cause welding of the sheared-off peak to the opposite high spot, a piece of transferred metal is formed. The intensity of the temperature flash depends on such conditions as speed, load, friction coefficient, and thermal conductivity. If the conditions are such that no appreciable diffusion can occur, the adhesive force, if sufficiently strong, can still keep the metal particle on the opposite high spot. The small piece of metal sheared from the high spot will leave as a loose wear particle when both diffusion and adhesion are feeble.

(NOTE: Detailed description of the experimental technique, the apparatus and the experimental results that support this theory is given in Ref. 1)

OTHER THEORIES. The Molecular Theory. In 1929, Tomlinson[7] proposed that friction is caused by the irreversible disturbance of molecules of the surfaces in contact. This molecular theory of friction was not well supported by the observed numerical values of the coefficients of friction. According to his idea, friction is entirely a surface phenomenon which is now known not to agree with observations.

Tomlinson's picture involves only disturbance of molecules on the surface without dislodging them from the surface or transferring them from one surface to another. Accordingly, friction will, under no circumstance, be accompanied by metal transfer or wear. Later, Holm[4] modified the molecular idea to explain wear without mentioning metal transfer. He suggested that, in sliding one high spot over another through a certain distance, an atom on one of the high spots will encounter a certain number of atoms on the opposite high spot and, from among those atoms encountered, a certain fraction will be dislodged and form wear product. Thus he arrived at the conclusion that wear is directly proportional to the normal load W and the distance of travel L and inversely proportional to the mean flow p_m. The experimental fact that loose wear particles are much larger than atomic size is not in agreement with Holm's idea. This basic drawback has already been pointed out by Burwell and Strange[8]. There are many other unsatisfactory phases of Holm's idea. For example, numerous experimental results indicate non-linear dependence of wear on the normal load. Another example is the incapability of Holm's theory in explaining the dependence of wear upon many factors other than W, L and p_m.

The "Welding" Theory. The "welding" theory that has been advanced by Holm[4] and Ernst and Merchant[9] and Bowden[10] is primarily for explaining the shear component of friction. Local "welding" between contacting surfaces establishes continuous metallic junctions which have to be sheared off during sliding, giving rise to the frictional resistance. A simple approximate relation between the normal load W and the area of actual contact is $A = w/p_m$, where p_m is the mean flow pressure of the high spots. The force F required to shear the metallic junctions is equal to the product of shear strength s and A. Thus friction coefficient μ is equal to s/p_m.

Friction, metal transfer and wear are very closely related so that only under special conditions will one of them occur without being accompanied by the others. The important assumption of the "welding" theory that continuous metallic junctions are always formed between two actually contacting high spots, causes difficulty in explaining the formation of the loose wear particles, which usually accompany friction. When one surface slides over another, weak metallic junctions will separate along the original interfaces yielding neither transferred metal nor loose wear particles, and strong junctions will cause only metal transfer. No loose wear particle could ever be formed by rubbing two dry surfaces!

DISCUSSION. A broad and detailed discussion covering all phases of metal transfer and wear is not intended, because it is impossible to make such an extensive discussion within a limited number of pages. The detailed analysis of the effect of various factors will be presented elsewhere in a series of publications. Discussion here will be confined to those factors which are relatively simple to explain and those which illustrate the difference of the various theories.

Shear Component of Sliding Friction and the "Normal Friction." Shear component is usually the

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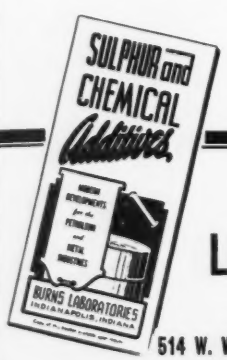

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major component of sliding friction. It is obvious that this new theory also explains the shear component of sliding friction. The difference between the "welding" theory and the new theory can be summarized as follows: The new theory explains metal transfer in the absence of adhesion as the consequence of frictional heat, whereas the "welding" theory suggests that "welding" is the cause of friction. The assumption about "welding" has been under question because no conclusive evidence has been produced demonstrating the "welded metallic junctions" between two metallic surfaces that are exposed to the ordinary atmosphere. It has been a controversial subject that, if welding is responsible for the sliding friction, there will exist, in separating two contacting surfaces in the normal direction, a normal frictional resistance of the same order of magnitude as the sliding friction. "Normal friction" between two perfectly clean surfaces in high vacuum has been observed. For example, Gwathmey, Leidheiser and Smith[11] have demonstrated that a force of the order of the normal load that had been used in pushing two oxide-free surfaces of copper single crystals together was required to separate the crystals in the normal direction. The reason of "welding" is metallic adhesion.

Contrary to the expectation, "normal friction" is usually not detected between surfaces exposed to the ordinary atmosphere. Gwathmey et al. observed no detectable "normal friction" between copper single crystals in air. A picture explaining the absence of "normal friction" by the presence of elastic recovery has been offered (Cf. Ref. 9). The explanation is not convincing. In the experiments of Gwathmey et al cited above, the elastic recovery of these delicate crystals is very feeble. It is more likely that the absence of "welding" between oxide-covered surfaces is responsible for the failure in detecting "normal friction".

In the presence of elastic recovery, "welding" or adhesion can still occur without being detected by "normal friction". If the elastic recovery is very weak, failure in detecting "normal friction" will indicate the absence of "welding" or adhesion. With this in mind, McFarlane and Tabor[12] chose for their experiments such soft metals as indium and lead where the elastic recovery is very small. They found that the "normal friction" between a steel ball and a lead surface that has been exposed to air for eight hours is zero. They seem to be of the opinion that there is some adhesion between oxides and then offer the explanation for the lower "normal friction" with thicker oxide film, that the thicker oxide layer could be assumed to be weaker inside the layer or could be assumed to attach less firmly to the underlying surface. Consequently, the absence of "normal friction" would indicate that no force would be required to split an oxide layer into two thinner layers or no force would be required to separate from the underlying surface the oxide layer that is formed on a lead surface after eight hours' exposure to air. Whether their explanation is reasonable or not is only a matter of secondary importance. The important conclusion that can be drawn from McFarlane and Tabor's data and the above discussion is that the absence of "normal friction" indicates no adhesion or "welding" between oxide-covered surfaces.

In the "welding" theory, adhesion or "welding"

comes in first and then friction follows. If there is no adhesion, there will be little friction. McFarlane and Tabor's result of no adhesion between oxide-covered surfaces gives rise to the difficulty of explaining the magnitude of static coefficient of friction between two oxide-covered surfaces under the condition of no adhesion.

Surface Contamination and Lubricant Film. Friction, metal transfer and wear between contaminated surfaces are important, because metallic surfaces in ordinary atmosphere are always covered with a layer of surface film. In the new theory, interlocking effect of plastic roughening is the primary cause. Consider surface films of thickness that is small as compared with the average depth of roughening. A thin, tenacious surface film may remain continuous on the roughened interface and prevent adhesion. Under such circumstances, no "normal friction" would be detected, whereas sliding friction is still principally the force required to shear the high spots interlocked by plastic roughening, because the interlocking effect is not noticeably weakened by the surface film. Consequently, no metal will be transferred, if frictional heat is not intense enough to cause appreciable diffusion. If the thin surface film is brittle, the roughening process can establish naked metal to naked metal contact at regions where metal of one high spot forces its way into the opponent high spot as represented by the dotted portion along the roughened interface in Fig. 2. If the degree of roughening is small, "normal friction" may not be detected because of elastic recovery. With large degree of plastic roughening, "normal friction" can become appreciable and metal

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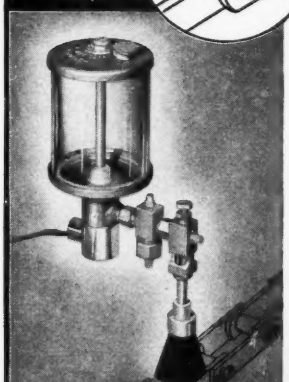

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transfer in the regions shown by dotted lines in Fig. 2 can occur during normal separation without sliding motion. The commercial "cold welding" technique employing excessive plastic roughening in joining two carefully cleaned metallic parts in air at room temperature is based on the same principle. The amount of plastic roughening in the commercial "cold welding" technique is far greater than that usually encountered in rubbing two surfaces together.

The presence of a surface film has little influence on the roughened interface in resisting tangential motion until it reaches a thickness comparable to or greater than the average depth of roughening. If the film is weaker, breakage will occur inside the surface film, being accompanied by smaller friction. The chance of metal transfer is greatly reduced, and wear takes place within the surface film. In the above discussion, the surface films are not intentionally introduced. Boundary lubrication is merely a case where a weak surface film is purposely added to the rubbing surface. Under unfavorable conditions, the average degree of roughening may exceed the thickness of the surface layer including the oil film on top of oxide and, thus, metal transfer and wear can still take place. When metal transfer and wear cause serious trouble, one can either introduce a thicker surface film to make shear take place inside the layer, e.g. by adding chemically active additives to the lubricant, or to reduce the roughening by reducing the load so that the average depth of roughening becomes smaller than the thickness of the surface film.

Surface Roughness of the High Spots. The effect of surface finish on friction, metal transfer and wear has also been a controversial subject. Neither the molecular theory nor the "welding" theory suggests any dependence on surface finish of rubbing parts. However, it is found in practice that the surface finish of the harder member of a pair of sliding surfaces is important with regard to the friction phenomena and the wear process. For example, a great deal of attention is usually paid to the surface finish of the hard shaft and far much less to the surface finish of the soft bearing.

Surface finish is really not a simple factor to discuss, because it influences the size and distribution of areas of actual contact as well as the surface roughness of the high spots. In order to simplify the present discussion, let us assume that the surface roughness of the high spots is directly related to the surface finish and neglect the effect of surface finish on the size and distribution of areas of actual contact. If two surfaces of the same metal having reasonable surface finish slide one over another, the original finish of the high spots has little influence because the interface is roughened by the process of plastic deformation, and it is the plastic roughening that causes interlocking. If the surfaces in contact are quite different in hardness, the surface finish of the hard member is important, for the soft high spots will deform plastically to fit more or less the original roughness of the hard high spots. The original surface roughness of the soft high spots will be obliterated by plastic deformation, and the interlocking effect depends largely on the original roughness of the hard high spots. The conclusions regarding the effect of the surface roughness are thus in good agreement with the practical experience.

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Coming Events

FEBRUARY

15-17 American Petroleum Institute (Lubrication Committee), Sheraton-Cadillac Hotel, Detroit, Mich.

15-17 Joint ASTM-TAPPI Committee on Petroleum Wax, Commodore Hotel, New York, N. Y.

23 Joint ASTM-AICE meeting (Southern California District), Los Angeles, Calif.

MARCH

1-5 American Society for Testing Materials (Spring meeting), Shoreham Hotel, Washington, D. C.

3-5 American Petroleum Institute (Div. of Production, Southwestern District), Rice Hotel, Houston, Tex.

8-10 American Institute of Chemical Engineers, Statler Hotel, Washington, D. C.

8-10 Oil Industry Information Committee, Shamrock Hotel, Houston, Tex.

17 Joint ASTM-ACS meeting (Philadelphia District), Hotel du Pont, Wilmington, Del..

24 to Apr. 1 American Chemical Society, Kansas City, Mo.

25 National Industrial Conference Board, Ambassador Hotel, Los Angeles, Calif.

APRIL

5-7 American Society of Lubrication Engineers (9th Annual Meeting & Exhibit), Netherland Plaza Hotel, Cincinnati, Ohio.

(Book Reviews, from page 19)

ASTM Committee D-2 on Petroleum Products & Lubricants held a symposium at their 1952 Meeting on the subject to present the views and latest work of leading experts in the field. Papers included are: Introduction, by DeVilliers; The Current Status of Fretting Corrosion, by Campbell; Fretting Corrosion Tendencies of Several Combinations of Materials, by McDowell; Influence of Fretting Corrosion on the Fatigue Strength of Fitted Members, by Horger; Effect of Lubricants in Minimizing Fretting Corrosion, by Herbek; and Test Equipment for Evaluating Fretting Corrosion, by Uhlig.

Standards on Petroleum Products (Nov. 1953)

by American Society for Testing Materials, Publishers, 1916 Race St., Philadelphia 3, Pa., 1953, 854 pages, price $6.00.

This special compilation of ASTM specifications, tests, and definitions covering petroleum products and lubricants is one of the most widely distributed ASTM publications. It provides in compact, readily usable form the widely used ASTM standards in this field. The 1953 edition gives in their latest approved form over 100 test methods, numerous specifications, lists of definitions of terms relating to petroleum and to rheological proper-

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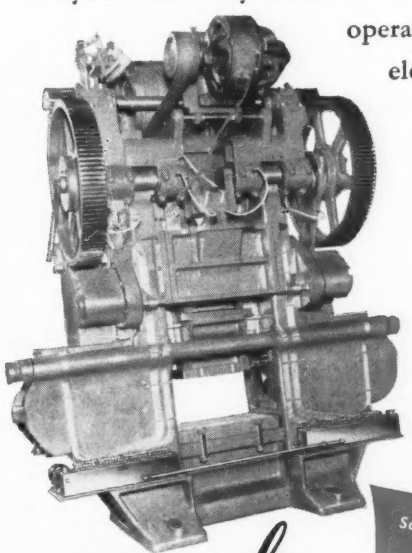

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ties of matter; and recommended practice for designating significant places in specified limiting values. (NOTE: The methods of test for knock rating of engine fuels are issued in a special volume and are not included in this compilation.)

(Section News, from page 15)

with C. W. Nichols, Jr., Socony-Vacuum Oil Co., Inc., as Director.

F. H. Stedman, Esso Standard Oil Co. (Co-author, F. D. Klein), presented a paper entitled Lubrication of Modern Aircraft, in which it was pointed out that the range of ambient temperatures over which aircraft greases and oils must give satisfactory performance is now approximately —100 F. to +300 F. Satisfactory greases and oils are generally available for most all applications required in modern civilian and military aircraft. However, future higher performance aircraft will require lubricants of higher load carrying ability and oxidation stability for satisfactory performance over even wider temperature ranges.

M. C. Hardin, Allison Division of General Motors, presented a paper entitled Lubrication of Aircraft Turbine Engines, in which the problems associated with the lubrication of various types of aircraft turbine engines were discussed. Material progress has been made through the use of synthetic lubricants, but it was emphasized that the currently available materials are not able to satisfy all present requirements and are expected to be more deficient in future engines. (Submitted by E. Landau, Treas.)

Pittsburgh October meeting—W. H. Millett, Carbide & Carbon Chemicals Co., presented a paper on the subject of Synthetic Lubricants.

November meeting—A "Stump the Experts" session with O. L. Brandes, Gulf Research & Development Corp., as Moderator, and Panelists M. D. Baker, West Penn Power Co.; J. Boyd, Westinghouse Research Lab.; A. O. Willey, The Lubrizol Corp.; and C. A. Bailey, U. S. Steel Corp. (Submitted by L. A. Tachoir, Sec'y-Treas.)

St. Louis September Meeting—Tour of the Lubricating Oil & Compounding Departments and Control Laboratory of Shell Oil Company's Wood River, Illinois, Refinery.

October meeting—A. McHugh, Ford Motor Co., discussed the objectives of laboratory and plant studies essential to the development of a sound plant hydraulic oil maintenance program. Continuous oil purification utilizing a filter as an integral part of the hydraulic oil circuit was indicated to provide optimum economic oil performance.

November meeting — A. J. Gruenewald, Lincoln Engineering Corp., presented a paper entitled Handling & Application of Lubricants, in which the advantages of automatic centralized lubrication in numerous installations were discussed in detail and illustrated by slides.

December meeting—K. L. Hulsing, General Motors Corp., presented a paper entitled Factors Influencing Operating Performance & Life of Diesel Engines. (Submitted by G. L. Stetson, Vice-Chrmn.)

Twin Cities December meeting—S. D. Craine, W. A. Jones Foundry & Machine Co., presented a paper entitled The Lubrication Engineer Services Industrial Gears, in which he pointed out that the majority of premature

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gear failures can be prevented by proper evaluation of periodic gear inspections. The meeting commenced with a technicolor sports film titled "Editors Afield." (Submitted by C. D. Johnson, Sec'y.)

Youngstown November meeting —A "Stump the Experts" session, with Panelists J. Lykins, Wheeling Steel Corp.; C. Lewis, Republic Steel Corp.; and D. Whitehead, Crucible Steel Co.

Traveling members of ASLE are welcome and invited to attend meetings held the second Thursday evening of each month at the VFW Building, Boardman St. (Submitted by J. A. Samuels, Sec'y.)

(Lube in the News, from page 23)

ucts in a fraction of the time now required. Using radio-active materials, by-products from the Atomic Energy commission's production, to find the percentage of hydrogen, or the ratio of hydrogen to carbon, in a compound, this new device may result in new specifications on such products as fuel oils, jet fuels, and other products where burning quality is particularly important. Analyzing a product with an accuracy of two-hundredths of one per cent by weight of hydrogen without changing the sample, the operator, who may be relatively unskilled, aims beta rays at the compound and measures how well they are absorbed. For further details, write: Standard Oil Co. (Ind.) (LE10/1), 30 E. 42nd St., New Chicago 80, Ill.

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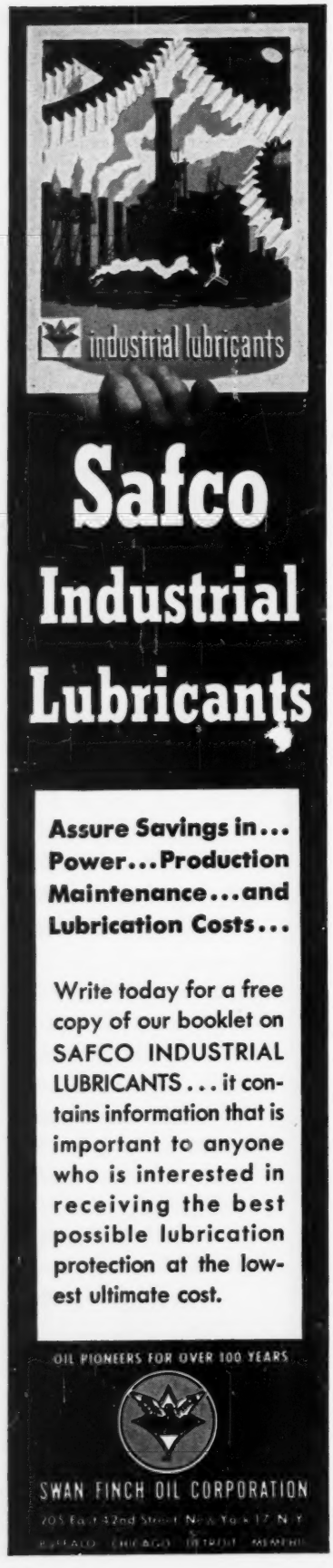
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Industrial Hydraulic Relief Valves Bulletin. A new bulletin covering a line of industrial hydraulic relief valves describes in detail the pilot operated valves which are available in four sizes, from ¼" to ¾". Built for long life, the valve has all the advantages of sub-plate mounting and allows full replacement of all working parts by the substitution of a new cartridge. Designed for operation over a temperature range from minus 65 F. to plus 160 F., with less than a 10% change in operating characteristics, it has a range of pressure setting from 350 to 5000 psi which is obtained by the use of three different quickly-interchangeable spring sets. Write for Bulletin No. AD, Pantex Manufacturing Corp. (LE10/1), Pawtucket, R. I.

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(Current Literature, from p. 33)

gine can be run on a 120-hour test procedure to provide a relatively simple and inexpensive method of comparing the performance of both straight and treated oils. Results so achieved line up well with service experience. The engine can be used to predict the performance of oils on the Caterpillar L.I. test with a reasonable degree of certainty. Reference is also made to the use of the AVI unit for testing production batches of heavy duty additives. Kerosene is used as a coolant. Anglamol Ltd. supported the development of the test procedure and carried out about 700 tests.

The Performance & Testing of Lubricating Greases under Wet Operating Conditions

by J. D. Smith & H. L. Hendricks, NLGI Spokesman, Vol. 17, No. 7, Oct. 1953, p. 23.

Little has been reported on the performance of greases under wet operating conditions, and, among the test methods for predicting the behavior of greases under these conditions, few have been standardized or gained wide acceptance. This paper discusses the

problem, outlines the behavior of conventional greases containing various thickening agents, and describes the available test methods. Several suggested new test methods are presented as well as the use of modifying additives to achieve greases with better performance characteristics under wet operating conditions.

Greases for Electric Motors
by E. G. Jackson & E. R. Booser, NLGI Spokesman, Vol. 17, No. 7, Oct. 1953, p. 20.

Values can be approximately assigned for bleeding rate, evaporation, and torque, which in turn depend on oil viscosity, soap type and soap concentration. Variations in the grease required by different types of bearings are recognized and can be specified. Some desirable properties are apparent, but improvement depends on further developments in rust inhibitors, oxidation resistance, etc. Properties which might provide an ideal grease for electric motors are proposed as a goal to work for.

Oxidized Petroleum Wax as a Grease Base
by J. C. Kirk & E. W. Nelson, NLGI Spokesman, Vol. 17, No. 7, Oct. 1953, pp. 16-17.

The mechanism of wax oxidation and some of the reaction variables are discussed. Several oxidation catalysts were investigated and a mixture of very finely divided manganese dioxide and zinc stearate was found to be particularly effective. Sodium, lithium, calcium, and mixed sodium-calcium greases were prepared. Results of fleet tests with a lithium grease are discussed.

The Computation of Viscosity & Diffusion in Liquid Suspensions & Compressed Gases
by E. U. Franck, Chemie Ingenieur Technik, Vol. 25, No. 8/9, Aug./Sept. 1953, pp. 442-446.

The equation derived by Maxwell & Rayleigh for the computation of the electrical conductance of binary heterogeneous alloys from the conductances of their constituents, is applied to the computation of viscosity (as impulse conductance) and of diffusion in heterogeneous binary systems. If the incorporated partner is highly diluted, this expression goes practically over in Einstein's viscosity expression. The new formula can be applied to the computation of the viscosity both of binary liquid suspensions (up to moderate concentrations) and of compressed gases (up to one-half of the density of the liquid.)

The Determination of the Density of Liquids
by G. Roeder, Chemie Ingenieur Technik, Vol. 25, No. 8/9, Aug./Sept. 1953, pp. 497-499.

The density of a liquid is determined at constant temperature and varying barometric pressures. If a correction is to be made for an air-free room, the correlation of pressure to the density of air is taken into consideration, whereas in general one reckons with a mean density of air. A Table for this correction is included; applicable for the 0.5-1.5 range of densities of liquids.

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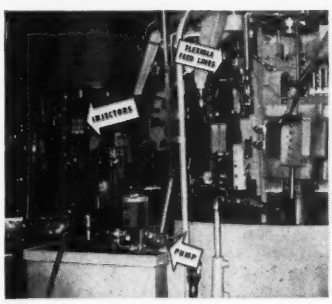

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Transition Flow of Fluids in Smooth Tubes
by V. E. Senecal & R. R. Rothfus, Chemical Engineering Progress, Vol. 49, No. 10, Oct. 1953, pp. 533-538.

Characteristics of laminar-turbulent transition were studied by measurement of point velocities and static pressure differentials in air flowing isothermally through two smooth tubes. Data were also obtained in the viscous and lower turbulent flow ranges. The presence of turbulence was observable at bulk Reynolds numbers greater than about 1,000. Major transition was marked by extreme changes in velocity profile.

Pour Point Depression of Lubricating Oils
by G. Gavlin, E. A. Swire & S. P. Jones, Industrial & Engineering Chemistry, Vol. 45, No. 10, Oct. 1953, pp. 2327-2335.

The studies reported in this paper were carried to establish the scope and potential utility of pour point depressing additives. They bear on the essential structural features of pour point depression. The occurrence of the pour point in wax-bearing mineral oils is attributed to the formation of a rigid network of solid crystalline components. The structure requirements of pour point depressants are indicated to be an alkyl side-chain length of at least 12 carbon atoms and a degree of polymerization commensurate with suitable over-all physical properties.

Friction & Consumption Characteristics of Motor Oils, Effect of Viscosity Index Improvers
by D. Frazier, A. R. Klingel & R. C. Tupa, Industrial & Engineering Chemistry, Vol. 45, No. 10, Oct. 1953, pp. 2336-2342.

Work reported here was undertaken to estimate quantitatively the effect of polymers in motor oils on consumption and engine friction. Paper discusses laboratory work on intercomparison of consumption rates of three "equal viscosity—equal viscosity index" motor oils prepared from suitable neutral stocks plus either of two commercial viscosity index improver polymers or a high viscosity index bright stock, and intercomparison of engine frictions of three "equal viscosity—equal viscosity index" motor oils, and intercomparison of 40 commercial motor oils of various listed properties.

Color-Clean Oil No Clue to Lubrication
by H. M. Gadebusch, SAE Journal, Vol. 61, No. 10, Oct. 1953, pp. 102-103.

The previously held idea that clear oil color is a sign of interior engine cleanliness and trouble-free lubrication is false. With an engine operated on straight mineral oil and without an oil filter, the contamination curve first rises in direct proportion to the operating time, passes the "color-clean" mark and then begins to level out gradually, thus indicating that sludge deposition has taken place at some point in the engine interior. After each successive oil change a much higher percentage of contamination can be measured than accountable for by the small amount of dirty oil remaining in the engine after draining. The bulk of the deposits has obviously remained in the engine to contaminate the new oil change.

Studies of the Formation of Carbons & Graphite
by S. R. Milliken, H. A. McKinstry & C. C. Wright, American Chemical Society, Abstracts, Sept. 6-11, 1953, p. 4K.

Carbon chars were prepared below 600 C. from pure organic compounds such as anthracene, diphenyl, benzoic acid, etc. An extensive x-ray study of the structural changes in these chars over the temperature range of 600° to 1000° and the subsequent changes in the conversion to graphitized or amorphous carbon when heated to 3000 C. has been made. A qualitative fundamental concept of the formation and nature of carbons and graphite is proposed.

The "Lube" Parade of Progress
by J. E. Dyer, American Petroleum Institute Preprints, Nov. 9, 1953, Meeting, 6 pp.

Recent advances made in the field of lubricating oil has radically changed the economics, the application, and the marketing of lubricants. A review of motor and industrial oils with respect to marketing shows the need for a reappraisal of the place and potential of lubricating oils among petroleum products.

Translated Tables of Contents of Russian Chemical Journals
by Anon., Chemical & Engineering News, Vol. 31, No. 44, Nov. 2, 1953, p. 4640.

Associated Technical Services will publish in 1954, on subscription basis, translated tables of contents of Russian chemical journals, with brief indicative abstracts. Russian journals it will cover are: Journal of Applied Chemistry, Progress in Chemistry, Bulletin of the Academy of Sciences of the U. S. S. R., Division of Chemical Sciences, and Ukranian Chemical Journal. Further details can be obtained from ATS, P.O. Box 271, East Orange, N. J.

The Adsorption of Benzene & Water Vapor by Molybdenum Disulfide
by E. V. Ballou & S. Ross, Journal of Physical Chemistry, Vol. 57, No. 7, Oct. 1953, pp. 653-657.

This investigation was undertaken to answer the fundamental question of whether the surface of molybdenum disulfide is, like that of graphite, nonpolar and hydrophobic. Adsorption isotherms of nitrogen, benzene and water vapor on molybdenum disulfide were determined. A hydrophilic layer of molybdenum trioxide is found on sulfide surfaces that have been heated to 110°. The oxide-free surface is found to be hydrophobic.

Cutting Fluids for Aluminum Bronzes
by Anon., Scientific Lubrication, Vol. 5, No. 10, Oct. 1953, pp. 27-28.

The addition of copper or zinc or both to aluminum increases the tensile strength considerably and also affects the thermal conductivity of the metal. For most ordinary turning operations a soluble oil yielding the normal milky emulsion will be satisfactory. The usual concentration is 1 part oil to 20 parts water (for grinding 1:40). Not all soluble oils are suitable owing to chemical attack. A satisfactory soluble

oil will be one where the emulsifier is based on the use of such non-fatty bodies as the petroleum sulphonates. For other operations, such as milling, on automatics, etc., where a straight cutting oil would be preferred, a thin, free-flowing mineral oil blend should be selected. Once again, the nature of the alloys will prevent the use of certain bodies, particularly the E.P. additives containing active-corrosive-sulphur.

Storage Stability of Oils Containing Pour Point Depressants under Fluctuating Temperature Condition

by Anon., ASTM Bulletin, No. 193, Oct. 1953, p. 24 (Statement of unsolved problem contributed by Committee D-2 on Petroleum Products & Lubricants).

The present ASTM method for cloud and pour point, though adequate for neutrals and bright stock without pour point depressants, is inadequate for unfiltered cylinder stocks and unrefined residuals which contain natural pour point depressants and for oils containing added synthetic pour point depressants. Recently the Esso laboratories developed several cycling temperature tests, now under study by ASTM Committee D-2. The pour point depressant is adsorbed on the wax precipitated and prevents wax crystals from growing. However the problem is not understood fully. A list of unanswered questions is given.

A New Technique for Adding Additives to Oil Whilst in Use

by Anon., Scientific Lubrication, Vol. 5, No. 10, Oct. 1953, pp. 23-29.

A new method of inhibiting lubricating oils against oxidation is being tested by the National Research Council of Canada at Ottawa. The method by which the inhibitor is introduced is via the oil filter or by attaching a piece of metal to the drain plug or inclusion in some other part of the oil system. The additive is not normally oil soluble and reacts only with the early stage oxidation products. It is thus impossible to supply an excess and it is, therefore, reported to be safe with inexperienced motorists or inexperienced garage mechanics. The oil is contacted with the additive in this way continuously and therefore keeps itself automatically "fortified."

(Patent Abstracts, from p. 28)

Chemieprodukte G.m.b.H., Leverkusen-Rheindorf, Germany

In the manufacture of lubricating greases, the process which comprises saponifying with caustic soda an unsaturated oleaginous material selected from the class consisting of animal fats and oils, adding a mineral oil in a quantity at least equal to the quantity of said unsaturated oleaginous material and about 2 to 4 per cent of beeswax, heating the mixture to a temperature sufficiently high to drive off the water present until the mixture can be filmed upon cooling, cooling the mass, grinding it, dissolving it in a liquid hydrocarbon solvent, aging the solution for about 24 hours and adding, about 2-5%, to a mineral oil which is to be transformed into a grease.

Grease Compositions, Patent #2,658,-869

by F. H. Stross, S. T. Abrams, and W. H. Peterson, assignors to Shell Development Co.

A lubricating composition comprising a major amount of a lubricating oil having dispersed therethrough in colloidally gelled form in an amount sufficient to thicken said oil, an amorphous gel comprising a mixture of a major component of the group consisting of silica and polyvalent metal silicates and a minor component of the group consisting of polyvalent metal oxides and polyvalent metal hydroxides, the amount of polyvalent metal being between 2% and 25% by weight of the inorganic constituents of said gel, and from about 5% to about 70% based on the weight of the gel, of a hydrophobic surface active agent absorbed on the surfaces of the gel.

Cutting Oil Composition, Patent #2,-658,870

by H. A. Ambrose and O. L. Brandes, assignors to Gulf Research & Development Co.

An improved cutting oil composition adapted to be applied as a thin high-speed jet consisting of at least 50% by weight of a mineral lubricating oil having a viscosity of about 70 to about 300 SUS at 100 F.; about 1 to about 20% by weight of a mineral lubricating oil having a viscosity between about 100 and about 200 SUS at 210 F.; about 0.5 to about 15% by weight of a fixed fatty oil; and about 1 to about 10% by weight of the residue which remains after removal of the solvent from the extract obtained by extracting a petroleum distillate oil with a solvent capable of removing polycyclic hydrocarbons and sulfur compounds from petroleum distillate oils, said residue having a viscosity between about 55 and about 70 SUS at 210 F.

Lubricating Composition, Patent #2,-658,871

by P. V. Smith, Jr. and J. B. McPherson, Jr., assignors to Standard Oil Development Co.

A lubricating composition which comprises a major amount of a mineral lubricating oil having combined therewith about 0.5 to 3.0% by weight of an ester of phosphoric acid, said ester being formed by reacting phosphorus oxychloride with the crude product obtained by subjecting a copolymer of propylene and butylene to the oxo process.

Composition for Manufacture of Sand Cores, Patent #2,659,654

by J. B. Tuttle, assignor to Standard Oil Development Co.

A core oil composition comprising about 30 to 50% by weight, based on the total composition, of drying oil, 15 to 35% of an unsaturated clay-filtered polymer of cracked gasoline derivation and 5 to 20% by weight of a viscous mineral oil of at least 100 and not more than 500 SSU viscosity of 210 F.

Lubricating Greases, Patent #2,659,-695

by J. Faust, assignor to L. Sonneborn Sons, Inc.

In the method for the production of lubricating grease by saponification the improvement which comprises intimately contacting a mineral lubricating oil,

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containing a solution of at least one member of the group consisting of fats and acids thereof, with an emulsion, said emulsion being a water-in-oil emulsion containing a petroleum mahogany sulfonate dissolved in a mineral oil, water in an amount to constitute the discontinuous phase of the water-in-oil emulsion and having an insoluble grease forming metal hydroxide dispersed throughout in amount sufficient to form a soap with said group member.

Lubricating Oil Containing a Metal Salt of a Condensation Product of an Alkyl Substituted Phenol and an Olefin Oxide, Patent #2,659,696
by L.L. Neff, assignor to Union Oil Co. of California
A mineral lubricating oil containing between about 0.05% and about 20% by weight of an oil-soluble metal salt of the acidic reaction product obtained by condensing, in the presence of an acid condensation catalyst, at a temperature between about —20 C. and about 200 C., between 0.5 and 3 mols of a hydrocarbon substituted phenol in which the hydrocarbon substituent contains between about 4 and about 30 carbon atoms with 1 mol of a low molecular weight olefin oxide containing between 2 and about 8 carbon atoms per molecule.

Lubricating Oils Containing an Acyloxyacetic Acid as a Rust Inhibitor, Patent #2,659,697
by S. J. Wayo, assignor to Sinclair Refining Co.
Lubricating oil compositions consisting essentially of a hydrocarbon mineral oil with a small amount, sufficient to impart rust preventing characteristics to said compositions, of an acyloxyacetic acid in which the total number of carbon atoms in the acyl group is at least 8.

Combination Process for Producing Lubricating Oils, Patent #2,660,552
by F. H. Blanding, assignor to Standard Oil Development Co.
The process of producing a lubricating oil comprising the steps of: catalytically cracking a gas oil feed stock boiling in the range of about 650 to 1100 F., fractionating an oil fraction from the cracked products boiling in the range of about 700 to 1100 F., and thereafter contacting said fraction with a selective solvent for aromatic compounds to remove aromatic compounds from the said fraction, said process including the step of dewaxing the said fraction after the catalytic cracking step, whereby a high viscosity index lubricating oil is obtained.

Lubricants, Patent #2,660,562
by W. N. Axe and W. B. Whitney, assignors to Phillips Petroleum Co.
A lubricating oil containing in solution from 0.1 to 25 weight per cent of an oil-soluble guanidine salt, of an organic sulfonic acid, selected from the group consisting of a synthetic alkyl aryl sulfonic acid and a petroleum sulfonic acid.

Trough Greases and Process of Preparing the Same, Patent #2,661,318
by F. H. MacLaren and L. P. Schwartz, assignors to Standard Oil Co.
A composition of matter consisting essentially of microcrystalline wax melting in the range of 120 F. to 165 F. and a refined mineral oil in such proportions that the oil will constitute about 50% to 85% of this mixture and added microcrystalline wax melting in the range of about 95 F. to 115 F. in an amount in the range of about 2 to 25 weight per cent based on the total mixture, both of said microcrystalline waxes being further characterized by having boiling ranges above 900 F. on a true boiling point basis.

Lubricating Composition, Patent #2,661,335
by J. M. Butler, assignor to Monsanto Chemical Co.
An improved grease which comprises a petroleum fraction containing from 1 to 20% of a fatty acid soap and from 0.01 to 5% of a compound made by the alkylation of a polystyrene having a molecular weight between 150,000 and 1,000,000, the alkyl group of said alkylated polystyrene having at least 4 carbon atoms and a total substituent having an average of from 6 to 12 carbon atoms per styrene ring.

Oil Compositions, Patent #2,662,055
by C. C. Towne, assignor to The Texas Co.
A substantially non-foaming and non-gas entraining composition of matter comprising a major portion of a liquid hydrocarbon oil and a minor proportion of an additive and a liquid dihydrocarbon silicone polymer of a minimum viscosity of about 900 cs.

Lubricating Compositions, Patent #2,662,056
by P. R. McCarthy, assignor to Gulf Research & Development Co.
A lubricant consisting essentially of a dispersion of a minor amount of water and a compound of a bentonite and an organic-nitrogen base in a major amount of a mineral oil, wherein the amount of said bentonite compound in said lubricant is sufficient to produce a composition having the consistency of a grease and wherein said water is present in an amount sufficient to increase the consistency of the lubricant.

Lubricating Compositions, Patent #2,662,057
by P. R. McCarthy, assignor to Gulf Research & Development Co.
An improved lubricant consisting essentially of a dispersion of a compound of a bentonite and an organic nitrogen base in a major amount of a mineral oil and a diaryl amine, wherein the amount of said bentonite compound in said lubricant is sufficient to produce a composition having the consistency of a grease and wherein the amount of said diaryl amine is sufficient to stabilize the lubricant against oxidational deterioration.

Lubricating Compositions, Patent #2,662,059
by P. R. McCarthy, assignor to Gulf Research & Development Co.
A lubricating composition consisting essentially of a dispersion of a compound of a bentonite and an organic nitrogen base in a major amount of an ester of an aliphatic dibasic acid, said ester having a majority of the properties of a hydrocarbon oil of lubricating grade, wherein said bentonite com-

pound is dispersed in said ester in an amount sufficient to produce a composition having the consistency of a grease.

Mineral Oil Compositions, Patent #2,662,856
by J. W. Bishop, assignor to Tide Water Associated Oil Co.
A composition comprising a mineral oil in major amount based on the weight of the composition and a combination of ingredients as defined hereinafter, one of said ingredients being an oil-soluble oil deterioration-inhibiting phosphorus- and sulfur-containing product of reaction between a terpene and a phosphorus sulfide and present in said composition in a small amount sufficient to inhibit deterioration of said oil and the other ingredient being an oil-soluble phosphorus and sulfur-containing metal salt of a product of reaction between a phosphorus sulfide and a hydrocarbon, said salt being characterized in that its metal component is an alkaline earth metal effective to provide a salt that inhibits the tendency of the aforesaid terpene reaction product to evolve H_2S and said salt being present in said composition in a small amount sufficient to provide in combination with said deterioration-inhibitor in said oil a composition inhibited against substantial evolution of hydrogen sulfide.

High Temperature Sodium Myristate Grease Containing N,N'Diphenyl Paraphenylene Diamine, Patent #2,663,690
by G. W. Eckert, assignor to The Texas Co.
A ball and roller bearing grease capable of lubricating anti-friction bearings at high rotative speeds for long periods at a temperature of 300 F. containing as the essential constituents a major proportion of an oleaginous liquid lubricating base, about 15-25% by weight based on the grease of a sodium soap of a soap-forming fatty material containing at least 70% based on the fatty material of myristic acid, said soap thickening said lubricating base to a buttery grease-like consistency, and about 0.5-8% by weight based on the grease of N,N' diphenyl para-phenylene diamine.

Grease Containing Diphenyl P-Phenylene Diamine and a Phosphorus Acid Ester and Method of Making Said Grease, Patent #2,663,691
by J. P. Dilworth, assignor to The Texas Co.
A lubricating grease composition comprising a lubricating oil, a metal soap of a fatty acid in grease forming proportion, a minor amount sufficient to impart thermal stability properties to the composition of N,N' diphenyl paraphenylene diamine added to the composition in solution in a minor amount of an oil soluble phosphorus acid ester, the latter serving as a carrier and a dispersing agent for the N,N' diphenyl para-phenylene diamine.

Lubricant and Method of Making Same, Patent #2,664,202
by J. D. Bartleson, assignor to The Standard Oil Co.
A method of improving the color of lubricants and lubricant additives which are reaction products of 5 to 60% by weight of a phosphorus sulfide and 95 to 40% of a mixture of one part by weight of an oxygen-containing organic compound reactive with phosphorus sulfide and selected from the group consisting of those compounds containing hydroxy, carbonyl, ether and carboxy radicals, and having a boiling point of at least 225 F. with 1 to 10 parts of a poly-olefin capable of improving the viscosity index characteristic of a lubricating oil, reacted together at a temperature in the range of 225 to 500 F., which method comprises treating such reaction products with an amount of adsorbent decolorizing agent in the range of 1 to 10% by weight of the reaction mixture and suffiicient to improve the color of the reaction product, said decolorizing agent having a water content in the range of 0 to 6% by weight, and recovering improved reaction product.

Coating and Lubricating Composition, Patent #2,664,399
by A. F. Kluender, assignor to Mor-Film Co.
A new composition of matter adapted to be dispersed in water and used to coat metal stock so as to form upon drying thereon an adherent protective film having rustproofing properties and which serves as a surface lubricant in working the metal stock, and consisting essentially of 20 to 30 parts by weight of a soap material selected from the group consisting of water-soluble medium titre soap having a titre value of from 22 to 30 and the alkali and free fatty acid-forming constituents of said soap, 34 to 40 parts by weight of $Na_2B_4O_7$ calculated on an anhydrous basis, and approximately 2 to 5 parts by weight of at least one member selected from the group consisting of trisodium phosphate, sodium nitrite and soda ash.

Reaction Products of an Octyl Acid Pyrosphosphate and Soyamine and Lubricants Containing the Same, Patent #2,664,400
by W. H. Woodstock and T. M. Beck, assignors to Victor Chemical Works
A lubricant for ferrous metals comprising a hydrocarbon oil and a rust-inhibiting quantity of an oil-soluble reaction product prepared by reacting a dioctyl acid ester of pyrophosphoric acid with 1 to 2.5 moles, per atom of phosphorus in the ester, of an amine composition containing at least 70% of unsaturated aliphatic primary amines each having at least 18 carbon atoms in the molecule until the reaction is substantially completed, the reaction being carried out at a temperature below that at which substantial decomposition of the reaction product occurs.

Lubricating Oil Composition of High Viscosity and Method for Preparing Same, Patent #2,664,436
by R. L. Heinrich, assignor, by mesne assignments, to Standard Oil Development Co.
A method for increasing the viscosity of an olefinic lubricating oil polymer having lubricating oil characteristics which comprise oxidizing the polymer at a temperature in the range from 300 to 600 F. in the presence of an oxidation promoter consisting of an amount no less than 0.1% by weight of quinoline to form an oxidized product having a substantially increased viscosity and a low neutralization value and recovering said product.

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Farval saves 50 minutes of each lubricating hour on Clearing press

FARVAL—*Studies in Centralized Lubrication* No. 150

GOOD news for cost-conscious management! Lubrication studies show that Farval saves 50 minutes out of every hour of oiling labor.

In a lubrication study of this Clearing press, hand-oilers were clocked with a grease gun as they lubricated, cleaned nipples, refilled gun, climbed and traveled from point to point. Averages show that it takes .747 minutes of labor per point to do all this. Thus, to grease this 41-point press by hand once per shift would take a little over 30 minutes. That's one hour a day based on a two-shift day.

Now, look at Farval. Experience shows that it takes only .188 minutes of labor per point to build up and hold pressure and lubricate. On the same basis, Farval lubrication takes a little over 10 minutes per two-shift day. This includes time for filling the Farval reservoir once every 5 days. In other words, Farval saves 50 minutes out of every hour of oiling labor! With larger Farval systems and with more points per press, the time savings are even greater.

Farval is the original Dualine system of centralized lubrication that delivers oil or grease under pressure to a group of bearings from one central station, in exact quantities, as often as desired. The Farval valve has only two moving parts—is simple, sure and foolproof, without springs, ball-checks or pinhole ports to cause trouble. Indicators at every bearing show that each valve has functioned.

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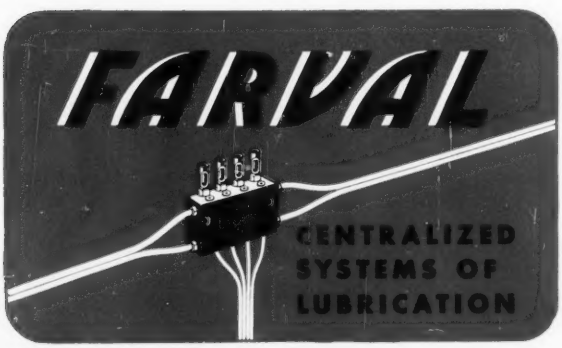

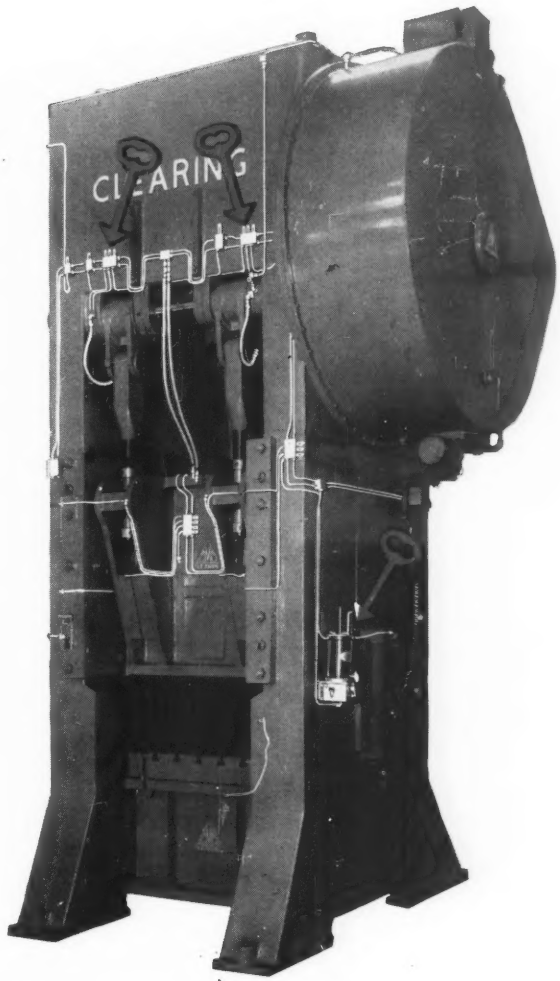

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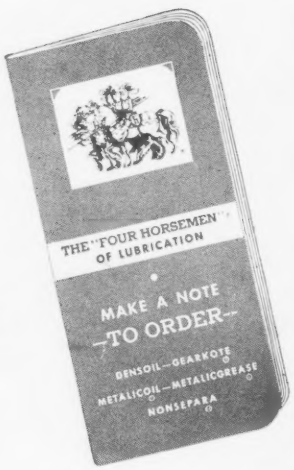

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